Low NOx Heavy Fuel Combustor Concept Program

Phase I: Combustion Technology Generation

Final Report

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NOTICE

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October 1981

CONTENTS

SUMMARY

The work described in this paper is part of the DOE/LeRC "Advanced
Conversion Technology Project" (ACT). The program is a multiple-contract
effort funded by the Department of Energy; technical program management
was provided by NASA LeRC.

The program objective was to conceptualize, design and demonstrate the
viability of low emission gas turbine combustors for utility and indus-
trial applications utilizing minimally processed petroleum residual and
coal-derived liquid fuels. Combustor configurations were constructed and
tested at the Westinghouse Waltz Mill site at full pressure of 12 atmos-
pheres (1.22 MPa) and a turbine inlet temperature of 1149°C (2100°F).
These configurations use rich-lean diffusion flames, rich-lean prevaporized/
premixed flames, rich-lean catalytic combustion, and lean catalytic
combustion to control the NO_x formation from fuel-bound nitrogen. The
test rig is of modular construction composed of a rich burner, quench
zone and lean combustion zone with fuel blending control and non-vitiated
compressed air control at each point of the three modules.

The configurations were tested on petroleum distillate fuel (ERBS fuel),
a coal-derived fuel (SRC II middle distillate), a petroleum residual fuel
and various blends of these fuels. NO_x, CO and UHC emissions were
measured, and other measurements were taken to evaluate combustor per-
formance.

Test results from this program showed that staged combustion (rich-lean)
with the venturi jet quench had the best overall performance in terms of
minimum NO_x emission for all fuels. The lowest value of NO_x is between
60 and 70 ppmv (corrected to 15 percent O_2) for ERBS fuel based on rich

primary equivalence ratios between 1.7 and 1.9. For SRC II fuel oil, the corrected NO_x emissions level is higher than the ERBS values by an increment of 70 to 85 ppmv, with conversion efficiency of 15 to 20 percent. SAE smoke numbers vary from 20 to 27. Quench module performance in limiting the NO_x-forming reactions is critical to the amount of thermal NO_x formed. Of the two quench modules tested, the venturi jet is more efficient than the vortex mixer.

Two types of lean burners were tested: a ceramic-lined section and a catalytic element. Results showed that the ceramic-lined section adequately completed combustion in all cases. However, the minimum NO_x level was independent of the primary equivalence ratio in the opposing swirl quench and the catalytic reactor. This fact is significant since smoke becomes a problem at higher equivalence ratios.

The test results of staged combustion (rich-lean) in Configurations 1, 2, 3 and 4 showed that NO_x emissions met EPA requirements for ERBS fuel and SRC II coal-derived liquid for a 30 percent cycle efficiency.

The catalytic combustor operating in the lean mode produced the least NO_x with ERBS fuel. Conversion rates occurred in the range of 40 to 50 percent for SRC II fuel oil with fuel-bound nitrogen at the 0.8 percent level. The corrected NO_x emissions for ERBS fuel is negligible in comparison with the NO_x expected for lean combustion in a conventional burner. Thus, the lean catalytic combustor produced ultra-low NO_x (one-half the EPA limit) on ERBS fuel.

The results for the Rolls-Royce combustor on ERBS fuel indicated that it is a low NO_x burner, and is an improvement in state-of-the-art combustor technology.

The combustor design tested on SRC II fuel oil converted about 50 percent of the fuel-bound nitrogen. The measured NO_x emission at the design point, for instance, was 280 ppmv. A smoke number of 7 (SAE) was observed for a CDL run; CO emissions ranged from 16 to 37 ppmv and the level of UHC was 2 ppmv.

The Rolls-Royce combustor achieved a low NO_x emission at the design condition on ERBS fuel. This value is comparable with that of the multiannular swirl burner. However, the Rolls-Royce combustor primary zone is not rich enough to prevent fuel-bound nitrogen conversion, which leads to conversion values of about 50 percent. The NO_x from the combustion SRC II was the highest of the combustors tested. These results showed that the Rolls-Royce combustor met the EPA regulation for ERBS fuel for a cycle efficiency of 30 percent.

The multiannular swirl burner was operated in the rich-lean mode for optimum conversion of the fuel-bound nitrogen. Maximum primary equivalence ratios were estimated to be about 1.5 with overall equivalence ratios of 0.2 to 0.25. These conditions were sufficient for low NO_x performance.

The NO_x results for SRC II fuel oil and residual oil showed an increase in NO_x in comparison with the results from the rich-lean combustor discussed above. It was apparent that the percentage of FBN conversion (15 to 30 percent at 0.8 percent FBN) is certainly lower than that of a conventional burner. Thus, the multiannular swirl burner also met the EPA requirement for ERBS and residual fuels. It appeared that the NO_x levels were comparable to the other rich-lean burners.

Results to date indicate that rich-lean diffusion flames can achieve high combustion efficiency with low fuel-bound nitrogen conversion.

Section 1
INTRODUCTION

Under Contract DEN 3-146, Westinghouse is participating in Phase I of the DOE/NASA low NO_x Heavy Fuel Combustor Concept Program as described by Lister, et al ([1]). The objective of this work is to develop low emission combustors for application to utility and industrial gas turbine systems that meet EPA standards of performance for stationary sources utilizing minimally processed petroleum residual and synthetic liquid fuels. Emphasis is placed on low NO_x production with a wide range of fuel-bound nitrogen contained in these fuels.

The combustors to be developed in this program will meet these requirements without injecting water or steam since these conventional techniques reduce thermal efficiency, are costly and sometimes ineffective.

PROGRAM DESCRIPTION

The low NO_x combustor program makes fundamental changes in the way in which fuels are burned in combustion turbines. The program goals reflect the EPA guidelines and utility standards of combustor operability over the engine load range, durability, maintainability, retrofitability, and performance in terms of efficiency, pressure loss, and uniformity of exit temperature profile.

A secondary goal of this program was to achieve ultra-low NO_x emissions (defined by NASA as 50 percent of the EPA limit) when operating on conventional petroleum distillate fuels. This objective is especially important in areas with high ambient air concentrations of NO_x. EPA emissions requirements may be satisfied by means of retrofit installations of combustors at existing sites in order to construct new fuel

burning installations for utility, industrial or cogeneration purposes. This report describes the work performed to develop low emission, fuel-flexible gas turbine combustors under these conditions.

The development work focused on four main areas:

- Conceptualization of advanced combustors

- Design of advanced combustors

- Fabrication of the combustion rig and combustors

- Demonstration of combustor viability by testing at current engine temperature and pressure conditions [1149°C/1.216 MPA (2100°F/12 atm)]

Minimally processed fuels such as coal-derived liquids contain fuel-bound nitrogen, which leads to an increase in NO_x emissions above those that are thermally produced. This conversion of fuel-bound nitrogen is high during normal engine combustion conditions as demonstrated in test results (2, 3). Therefore, the goal of the advanced combustor designs is to minimize the conversion of fuel-bound nitrogen to NO_x as well as thermally produced NO_x.

Combustor configurations selected for development and design include staged combustion (rich-lean) while utilizing diffusion flames, rich-lean prevaporized/premix flames, and catalytic models. The configurations have been fabricated and tested at full pressure and temperature. The test rig designed and constructed for this program was of modular construction for staged combustion. It consists of a rich burner module, a quench module, and a lean combustion module.

It also included on-line fuel blending control and a non-vitiated compressed air supply at each state point of the three modules.

Test results were obtained for petroleum distillate fuel, a coal-derived liquid (SRC-II middle distillate), and a petroleum residual fuel. Other results were obtained with various blends of these fuels. Emission measurements and operational performance data were obtained to evaluate

these combustor configurations. Analyses and evaluations of combustor performance was the basis for demonstrating the feasibility of the various low emission gas turbine combustor configurations.

EMISSION GOALS

The emission goals of this program are contained in the EPA NO_x Emission Standards, and are presented in Tables 1 and 2. In addition to NO_x emissions, CO, CO_2 and UHC also were monitored during the test program.

Table 1-1

LOW NO$_x$ HEAVY FUELS COMBUSTOR CONCEPT PROGRAM
SUMMARY OF GOALS AND OBJECTIVES

Emissions Limits (All Operating Conditions)

    Nitrogen Oxides   -   75 ppm @ 15% O$_2$ (1)

    Sulfur Dioxide   -   150 ppm @ 15% O$_2$ (2)

    Smoke   -   SAE No. 20

Performance Specifications

    Combustion Efficiency   > 99% at all operating conditions

    Total Pressure Loss   < 6% at base load power

    Outlet Temperature   > Equivalent to typical production
                                       engine combustor values

    Combustor Exit   < 0.25 at base load power
    Temperature Profile

General

    Retrofitable to current production and field engines

    Maintainable

    Fuel-Flexible - Capable of meeting emissions and performance
                    specifications on liquid fuels including petroleum
                    distillates and residuals, and synfuels from coal
                    and shale

Notes: (1)   These limits are subject to the constraints and corrections
               in the EPA Proposed Rule for Stationary Gas Turbines, Federal
               Register, 40 CFR Part 60 pp. 53782-53796, October 3, 1977,
               which is hereby incorporated by reference. This rule was
               published as a final rule on September 10, 1979. See Table 2.

        (2)   Goal represents practical limit with fuel sulfur content of
               approximately 0.8 percent, based on total conversion.

Table 1-2

EPA NOx EMISSION STANDARDS WITH 15 PERCENT OXYGEN IN
EXHAUST AS A FUNCTION OF CYCLE EFFICIENCY
FOR LARGE COMBUSTION TURBINES (>10000HP)

Cycle Efficiency	Metropolitan Statistical Area		Non-metropolitan Statistical Area**	
	Maximum without FBN	Maximum with FBN*	Maximum without FBN	Maximum with FBN*
25	75 ppm	125 ppm	150 ppm	200 ppm
30	90	140	180	230
35	105	155	210	260
40	120	170	240	290
45	135	185	270	320

* Assumes Fuel Bound Nitrogen (FBN) content equals or exceeds 0.25 percent.
** Gas or oil transportation and production only.

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Section 2
COMBUSTION TURBINE EMISSION CONTROL CONCEPTS

The combustion process in the modern industrial turbine is becoming more complex. For instance, there are many possible process paths for combustion of the fuel on which a turbine runs. In addition to the increased complexity of the combustion process, it has become the object of restraints. They focus particularly on reducing nitrogen oxides (NO_x) emissions.

Although alternative burner geometries may limit NO_x production, certain combustion process paths can achieve a similar result. This discussion concentrates on how these processes may be used to limit or eliminate NO_x production in modern gas turbine combustion equipment.

2.1 GENERAL CONSIDERATIONS FOR RICH/LEAN COMBUSTION

The metallurgical and cooling design of the present industrial combustion turbine restricts the usable gas temperature to approximately 1200°C (2200°F). This temperature corresponds to a required fuel/air ratio of less than 0.03 (kg of fuel per kg. of air). Conversely, the combustible range of hydrocarbon-air mixtures varies from a lean fuel/air ratio limit of about 0.035 to a rich limit of 0.20; the exact values are dependent on the temperature of the unburned mixture. Therefore, the range of fuel/air mixtures required for the gas turbine is outside the flammability range of hydrocarbons.

Although desirable, the combustion chamber cannot operate on a purely homogeneous mixture of fuel and air. The fuel/air ratio must therefore,

be reduced to combustible limits. This may be accomplished by two of the more obvious combustion processes.

One is to split the air supply into two parts, primary air and secondary air. Primary air is mixed with the fuel to form a homogeneous mixture; secondary air is used as the coolant for the burned primary mixture. The other method is to introduce the fuel into the entire air supply and burn it locally before a homogeneous mixture is formed.

The first process requires a combustor with the following components:

- An air divider to separate the primary and secondary air

- A fuel-injection and mixing zone where a homogeneous combustible mixture is formed

- A combustion zone where the combustion reaction occurs

- An additional mixing zone to recombine the burned mixture with the secondary air

In addition to these components, if a wide combustor outlet temperature range is desired, the primary air must be metered to maintain combustible mixtures. These conditions indicate that a split air stream, with combustion occurring in a primary homogeneous mixture, has many disadvantages. In terms of design and control, it is more complex in comparison to the popular heterogeneous combustion chamber, which minimizes cost and complexity by using a single fuel control. In a heterogenous combustor design, space and weight limitations are more easily accommodated.

Ideally, the heterogeneous combustion chamber progressively burns fuel in an air stream, thereby minimizing the temperature. Liquid or gaseous fuel is introduced in a low velocity region of the air stream, such as the wake of a bluff body or the dome end of a cannular burner; and partial mixing occurs due to turbulence. Then, on ignition, partial combustion takes place and produces a flame, which activates the burner.

Since insufficient time is available for complete mixing, the variation in the existing fuel/air ratio is great enough to ensure a number of regions in the flame tube that are within the flammability range. Once the burner is operating, the liquid fuel spray vaporizes progressively along the length of the flame tube. The slight amount of fuel vaporized in the vicinity of the fuel nozzle mixes with the incoming air and maintains a piloting flame, which is ignited by a spark from a hot wire.

As the fuel spray progresses along the flame tube, it is vaporized and mixed with the air coming into the flame tube from the air holes; the mixture is burned as it is formed. This process continues until all the fuel is burned.

After the combustion phase is complete, the remainder of the air coming through the holes acts as a diluent and reduces the temperature of the burned gases. In this type of burner, the formation and burning of the mixture occur simultaneously, and the ignition (or piloting source) is an inherent characteristic of the low velocity region in the upstream end.

Unfortunately, the time consumed by the evaporation of the fuel droplets and the simultaneous burning of the fuel in the vapor shell surrounding the droplet is conducive to NO_x formation. This combustion process produces thermal NO_x when the temperature level is high enough to produce the overall reaction at a measurable rate. This reaction is common to all combustion processes and is independent of the fuel.

Fuel-bound nitrogen, in contrast to air-bound nitrogen, forms NO_x-type compounds in the combustion process as fuel carbon forms CO and CO_2, and as fuel hydrogen forms H_2O. In this case, the nitrogen competes with the other elements in the fuel for available oxygen from the air.

The two types of nitrogen reactions are expressed by the following equations. In the first set of reaction equations, pyridine (a nitrogen-containing fuel) reacts with stoichiometric and sub-stoichiometric amounts of air.

Pyridine + Air = Products of Combustion

$$C_5H_5N + 6.75 (O_2 + 3.773 N_2) = 5CO2 + 2.5 H_2O + NO + 25.4 N_2$$
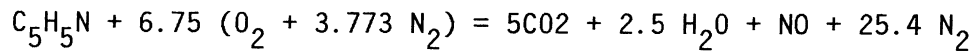
Pyridine + Insufficient Air = Products of Combustion

$$C_5H_5N + 3.75 (O_2 + 3.773 N_2) = 5CO + 2.5 H_2O + \frac{1}{2} N_2 + 14.1 N_2$$
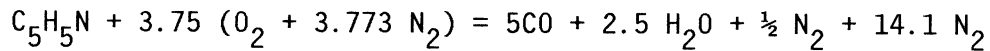
The interesting aspect of the last two equations is that fuel-bound nitrogen at rich conditions does not form its oxide, but is reduced to molecular nitrogen. The thermal reaction forming NO_x is usually characterized by the following two equations:

Thermal Reaction

$$N_2 + O_2 \rightarrow 2NO$$
$$2NO + O_2 \leftarrow 2NO_2$$
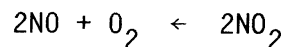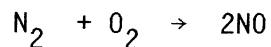
Differences in the appearance of the two types of nitrogen oxides depend on time and temperature of the combustion process. Fuel-bound nitrogen compounds appear simultaneously with CO and H_2; the formation of NO_x from the oxidizer appears later in the combustion process and is governed by a kinetic rate mechanism known as the Zeldovich mechanism ([1]). Even if this mechanism or more complex computer codes are used to predict NO_x formation, there are uncertainties in NO_x prediction due to a wide variation in the kinetic data ([3-7]).

With respect to a heterogenous mixture, thermal NO_x formation can be minimized by limiting the temperature of the reaction (very lean combustion), which limits the reaction rate of NO_x formation. However, fuel-bound nitrogen, when burned with excess oxygen, will produce some NO_x immediately (with the thermal NO_x forming subsequently) if thermal conditions are favorable.

In principle, NO_x formation can be limited by two processes:

- Low temperature combustion where the amount of NOx generated from the fuel-bound nitrogen is tolerable because thermal NO_x is minimized.

- Rich combustion in which oxygen is deficient and the relatively inactive nitrogen cannot compete with carbon and hydrogen for the oxygen. The application of this process to practical systems follows.

2.2 RELATIONSHIP BETWEEN THE COMBUSTION PROCESS AND NO_x FORMATION

Figure 2-1 is a plot of calculated flame temperature and the <u>air</u> equivalence ratio, which is a ratio of actual air-to-fuel mixtures to stoichiometric conditions. Using <u>air</u> equivalence ratios, lean mixtures will have a value greater than 1.0. In other sections of this report, a <u>fuel</u> equivalence ratio will be used; this is the reciprocal of the air equivalence ratio in which lean mixtures will have a value less than 1.0. Also shown (Figure 2-1) is the flammability range for the given combustor inlet conditions and the desired turbine inlet temperature. This range falls outside of the conventional flammability range and is obtained through a combustor with a split air system.

In Figure 2-2, the equilibrium NO_x concentration is shown as a function of the air equivalence ratio. These values are obtained from a computer
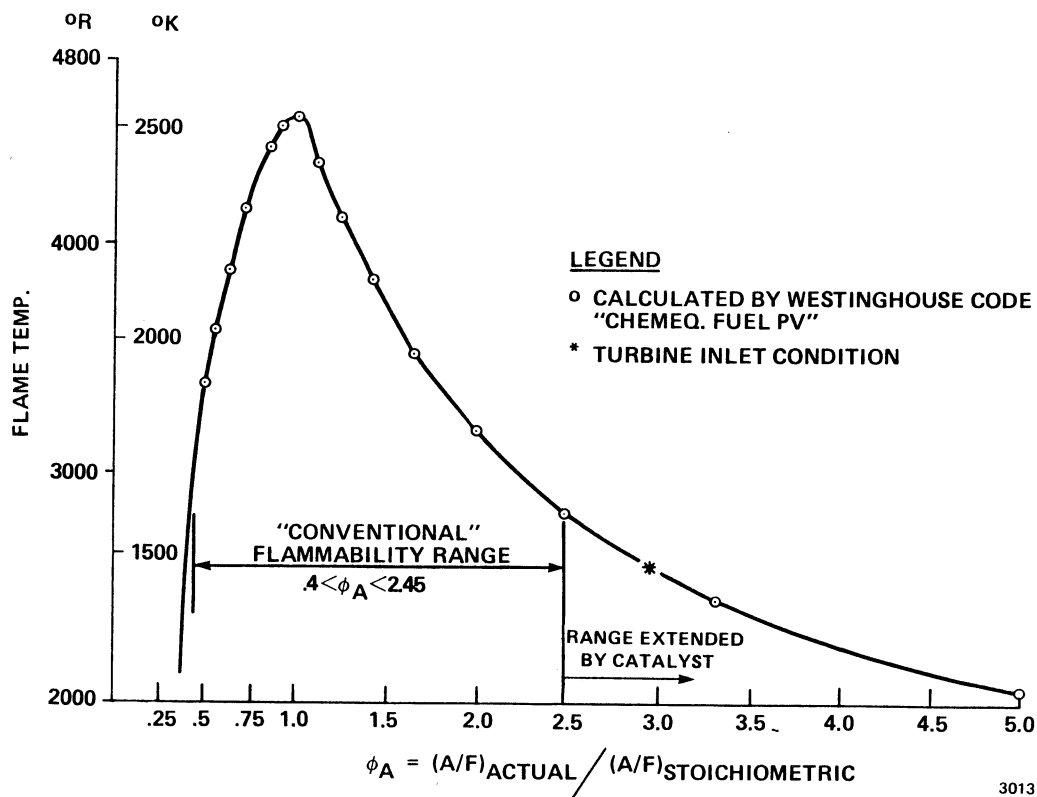

Figure 2-1.   Flame Temperatures as a Function of Air/Fuel Ratio With Indicated Flammability Range
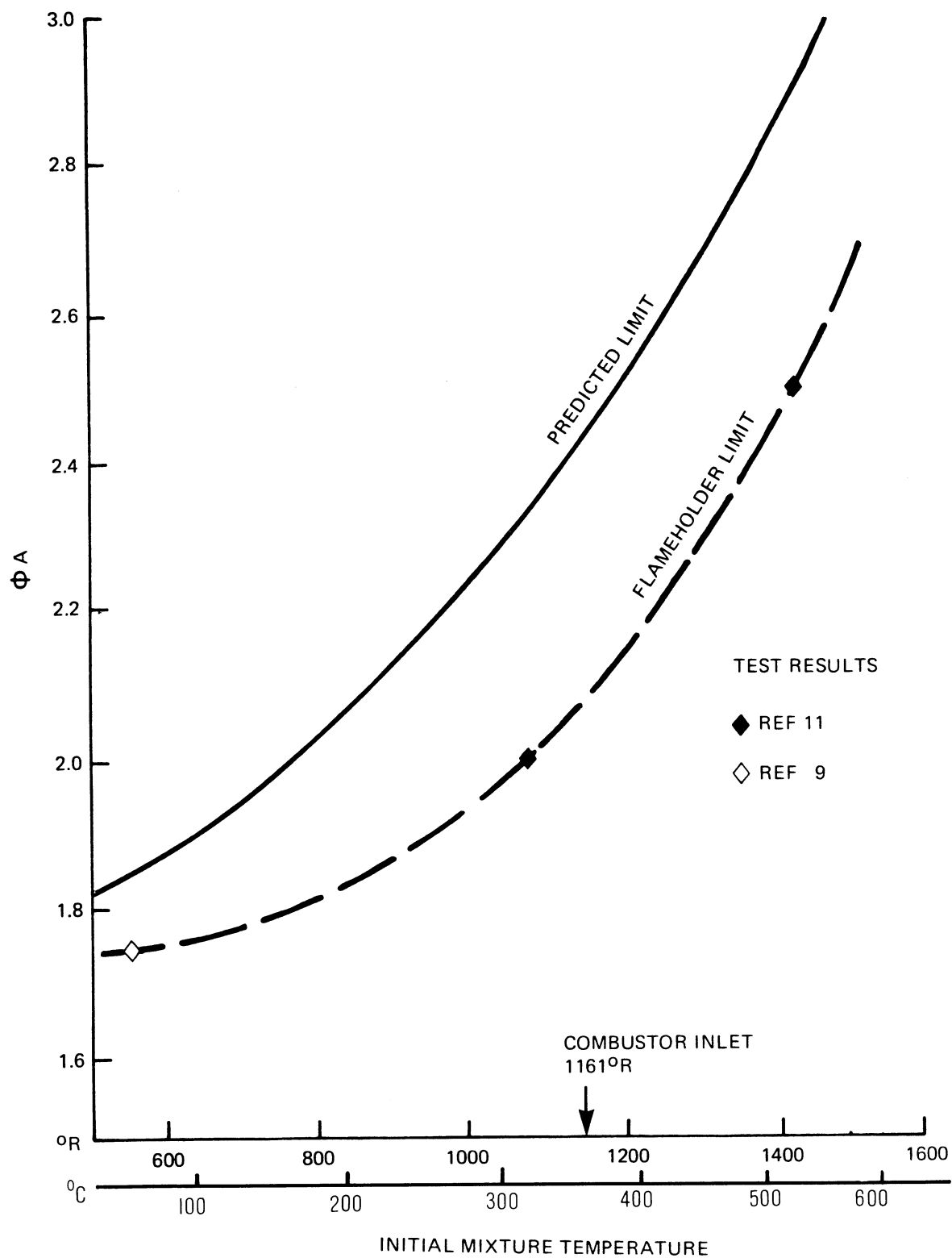
Figure 2-2.   Equilibrium Concentration of NO_x as a Function of Air Equivalence Ratio

code that calculates the equilibrium temperatures of the flame by mini-
mizing the free energy of the system. Fuel-bound nitrogen has an effect
on the equilibrium NO_x concentration with rich mixtures; however, the
equilibrium NO_x is insensitive to fuel-bound nitrogen in lean mixtures.
Also shown (Figure 2-2) are very low equilibrium NO_x concentrations
expected when a burner operates in a rich mode.

The rich/lean combustion process presents two basic problems. The first
involves the rich burner thermodynamic equilibrium. As it moves to this
level, the combustion process can generate NO_x values above the equilibrium
value. This condition can occur in a poorly mixed fuel/vapor-air mixture,
or in a droplet diffusion flame. The second problem involves the combustion
process from the exit of the rich burner to the turbine inlet. At this
point, additional air from the rich burner is supplied to the effluent.
This must be done so that the combustion process is quickly completed at
conditions that are unfavorable to the generation of NO_x. Since primary
combustion occurs at fuel-rich conditions and the final overall ratio is
lean, the term rich/lean is used to describe this type of combustion process.

The combustion process, particularly in the lean range, is much faster
than the production of NO_x. (An exception may be the burnout of CO.)
NO_x production from fuel-bound nitrogen, while not clearly understood
(9), seems to resemble the products of hydrocarbon combustion. The
conversion fraction of fuel-bound nitrogen to NO_x varies considerably
depending on fuel type and application (10,11). A burner operating in
the lean region at a lower temperature produces a small amount of thermal
NO_x. The exact amount depends on the residence time of the mixture and
the initial NO_x concentration in the combustion products from a lean
flame. Figure 2-3 shows the performance of a lean burner 0.457 M long
operating at a reference velocity of 30.5 M/S (100 fps), followed by a
dilution zone in which the final value of the equivalence ratio for the
required turbine inlet temperature is attained.

The NO_x concentration is shown in two ways: the primary zone concentra-
tion corresponding to the primary zone fuel/air ratio, and the burner
exhaust concentration where the primary zone concentration is diluted to

the final value. The flame temperature is also plotted as a function of
the equivalence ratio. These calculations were obtained from the num-
erical integration of the Zeldovich equation ($\underline{1}$) by means of a Runge-
Kutta procedure. The NO_x concentration in the primary zone (Figure 2-3)
will be less than 30 ppm if the maximum temperature is restricted to
values below 1725°C (3600°R). Final dilution with secondary air reduces
the NO_x concentration to less than 10 ppm.
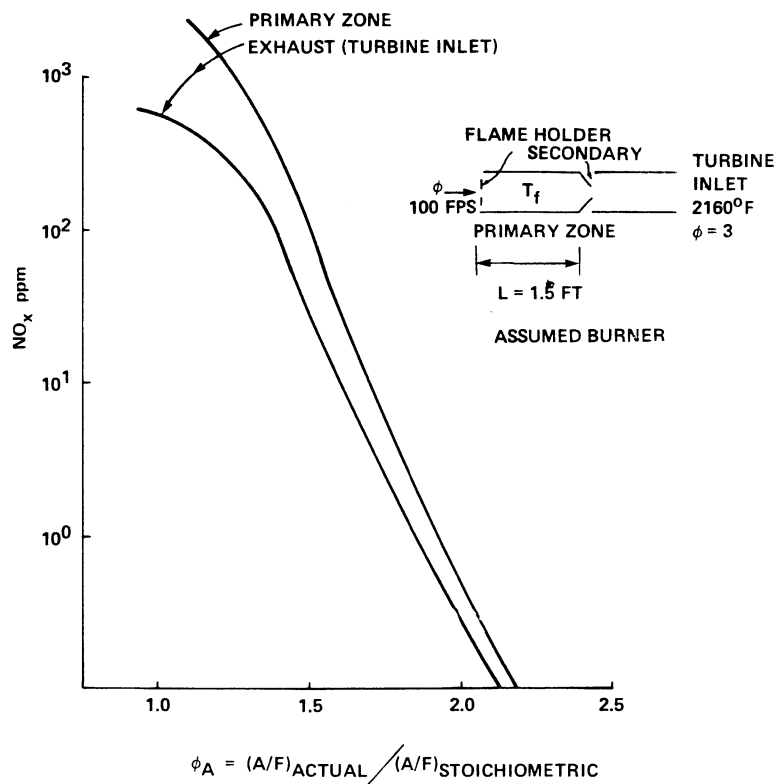
Figure 2-3. NO_x Generation for Lean Burner

Using a similar NO$_x$ standard of approximately 30 ppm for an equilibrated rich burner, a temperature of 1975°C (4050°R) is obtained using Figure 2-2 for the equivalence ratio of 0.65, and Figure 2-1 for the temperature at Φ = 0.65. There is a "window" on the temperature-equivalence ratio plot which represents the range in which minimal NO$_x$ can be expected. This window, shown in Figure 2-4, lies within the area of ABCD. Segment AB represents the lean limit of combustion. Segments AD and CB represent thermodynamic equilibrium conditions. Segment CD connects maximum acceptable NO$_x$ values on the rich and lean sides. This window represents the domain in which low NO$_x$-producing combustion processes can operate with final dilution to the required turbine inlet temperature condition ending at E. Any processes that occur below segment AB are only unreactive mixing processes in homogeneous burning.
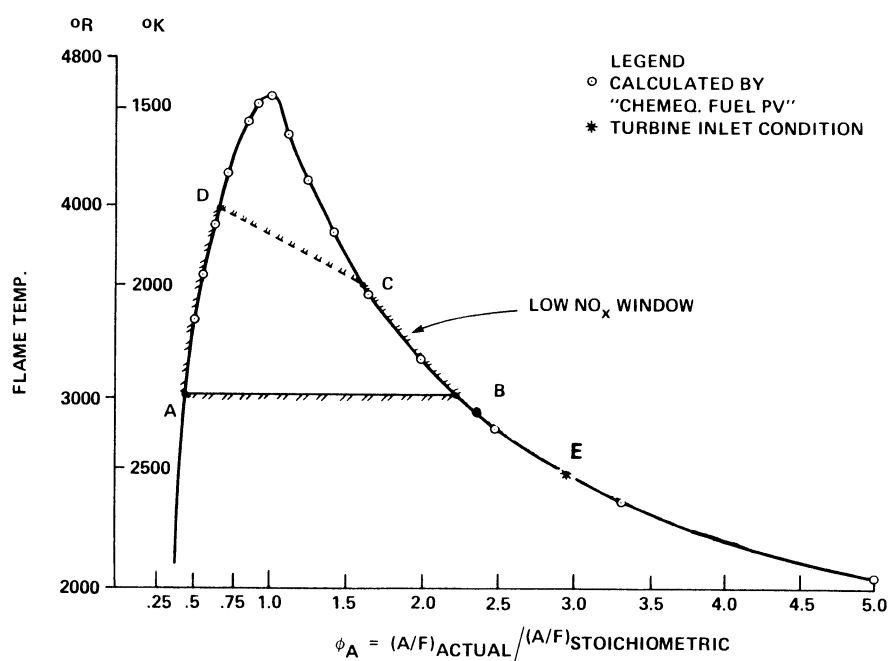
Figure 2-4.   Low NO$_x$ Window in Temperature Equivalence Ratio Plane

Lean combustion devices rapidly attain equilibrium for the products of
combustion of the hydrocarbons and fuel-bound nitrogen. In this case,
subsequent thermal NO_x generation is a function of the residence time in
the primary section. The residence time is a compromise between the
carbon monoxide burnout time and the thermal NO_x generation time. The
process proceeds along line CB and ends at E (Figure 2-4).

In order to use nitrogen-containing fuels, the concept of burning to
equilibrium conditions in the rich range is attractive. It is evident
(Figure 2-2) that the effects of nitrogen in the fuel can be maintained
at a low value in the rich zone for all fuels. However, one problem
remains. There must be a means by which discharge from a rich burner can
be mixed and processed to attain the required turbine inlet temperatures.

Figure 2-5 shows the low NO_x window with three idealized lines in addition
to the original equilibrium line. These lines represent a possible
combustion process starting from the discharge of an equilibrated rich
burner at point Y to the turbine inlet conditions at point E.

Point Y represents an equilibrated rich mixture from the rich stage of
combustion. This mixture is well within the allowable NO_x generation
range of less than 30 ppm (which on dilution to final turbine inlet
conditions will be in the range of 5 ppm if no further reaction occurred).
If the process is slowed by adding air to increase the equivalence ratio
from 0.5 at Y to 3 at E, the temperature would follow the equilibrium
curve YDLCBE ending with the equilibrium NO_x concentration of 820 ppm
(Figure 2-2).

If the process occurs more rapidly, the process line might be represented
by YMBE. Equilibrium temperatures will not occur and the NO_x concentra-
tion will be reduced. Since the process occurs within a finite time
interval, the concentration of NO_x is considerably less than equilibrium
as the NO_x concentration generation is kinetically limited by reduced
time and temperature.
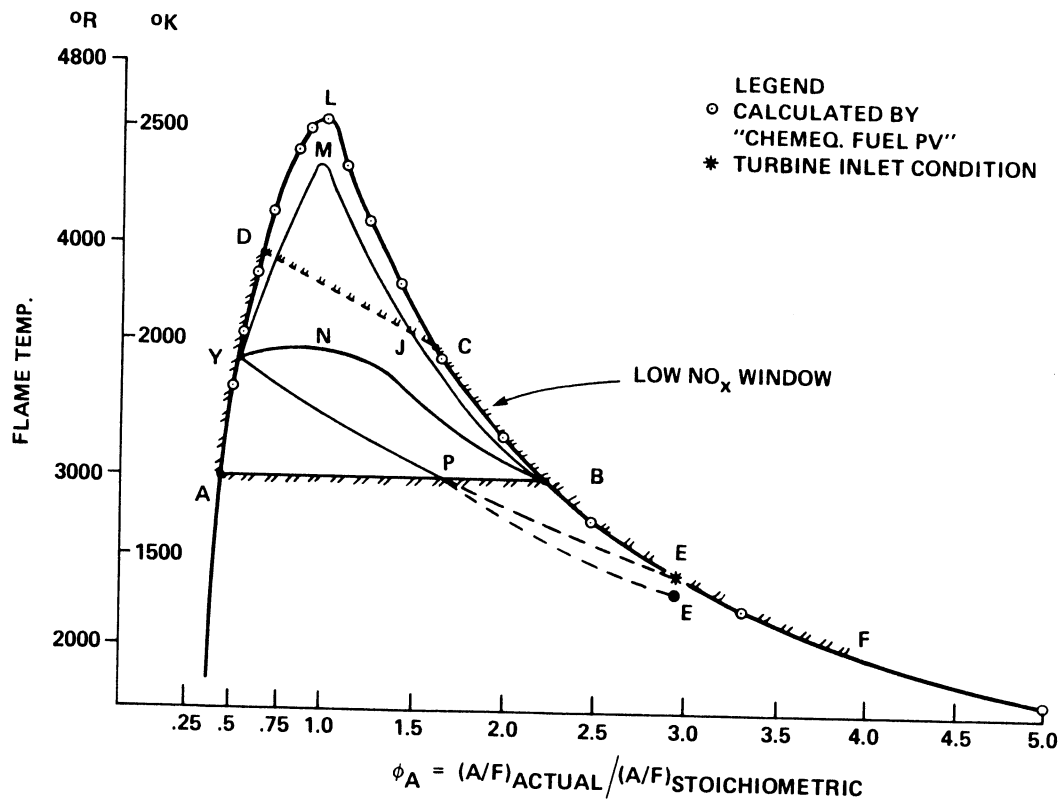
Figure 2-5.   Rich to Lean Process Lines

If the process line exceeds the allowable window, excessive NO_x may still
be generated. In fact, the NO_x concentration exceeds the equilibrium
value at E if an unfavorable kinetic situation prevents a reduction in
peak NO_x values attained near the stoichiometric peak. Any situation
that freezes any intermittent specie concentration between near stoich-
iometric and end conditions can result in NO_x concentration greater than
the end point equilibrium value of 820 ppm.

A process which reflects line YNBE is ideal since it remains within the
low NO_x window. From a physical viewpoint, such a process may be dif-
ficult to obtain since the cooling resulting from the mixing process, by
which the products of combustion and additional air are combined, must be
offset by the heating process of the reaction of the unburned species.

A process which reflects line YPE is extinguished at point P where the temperature is too low to sustain the process. As a result, re-ignition must occur between P and the equivalence ratio at B if point E is to be attained. Without a chemical reaction, the line beyond P cannot intersect with E. This incomplete process is shown as a dashed line. The process line below E is a pure mixing line with a lower temperature at the same equivalence ratio as point E. It represents a prematurely quenched condition.

This example illustrates the planned method of developing a rich/lean combustor. The rich section of the combustion chamber must produce an equilibrated mixture of hot gases capable of further burning. The competition between hydrogen and carbon for the limited oxygen leaves the nitrogen with essentially no available oxygen - thus, NO_x is very low. The combustion process must be rapidly completed at a low temperature to minimize any additional NO_x formation. Individual comustor designs developed to follow this process will be discussed in the next section.

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Section 3

COMBUSTOR DESIGNS

To meet the objectives of a low NO_x burner, advanced combustor designs
with fuel flexibility are divided into two major categories:

- Staged combustion to effectively control the fuel-nitrogen
  conversion to NOx in nitrogen bearing fuels

- Lean combustion for petroleum distillates or fuels with moder-
  ate fuel-bound nitrogen levels

In staged combustion fuel nitrogen is converted to N_2 in the rich stage
(primary combustion) with a minimum of conversion in the lean stage
(secondary combustion). The temperature in the lean stage is low enough
to minimize thermal NO_x formation.

Most of the combustors that were used to test this process have a modular
design; that is, burner components can be readily interchanged. Each of
the burner modules is housed in a pressure chamber and connected by
flanges. The modules are as follows:

- Rich burner module with a conventional head end diffusion flame
  or a premixed flame with an ignition source

- Quench module where additional air is introduced to reduce the
  gas temperature and chemical reaction

- Lean burner module where the combustion process is completed

- A dilution section where additional air is introduced into the
  existing fuel/air mixture in order to bring it to the desired
  turbine inlet temperature

In addition to the modular burners, this program used two combustors in which the individual aerodynamic and combustion processes are performed in more a integrated fashion.

The following combustor configurations were considered in this program:

Direct Injection - Rich/Lean

1. Direct Injection, Venturi Quench
2. Direct Injection, Vortex Quench
3. Direct Injecton, Vortex Quench, Perforated Plate
4. Direct Injection, Vortex Quench, Catalyst
9. Multiannular Swirl Burner
11. Rolls-Royce Combustor

Premix Rich/Lean

5. Recirculating Counter Swirl, Venturi Quench
10. Perforated Plate, Venturi Quench

Rich Primary Catalytic - Lean Staged Combuston

7. Catalyst A, Venturi Quench
8. Catalyst B, Venturi Quench

Rich Hybrid Premix/Direct Injection

6. Hybrid Piloted Rich Burner, Venturi Quench

Lean Catalytic

12. Catalytic

Lean Hybrid Premix/Direct Injection

13. Hybrid Piloted Lean Burner

The following description of the various design configurations begins with the modular designs, and continues through and ends with the integrated burners.

3.1 DIRECT INJECTION, VENTURI QUENCH - CONFIGURATION 1

Configuration 1, shown in Figure 3-1, consists of a rich burner, a
venturi-type quench section, a straight pipe lean burner with a separate
igniter, and a dilution section. Since fuel is introduced directly into
the rich burner, there is no separate fuel preparation module.
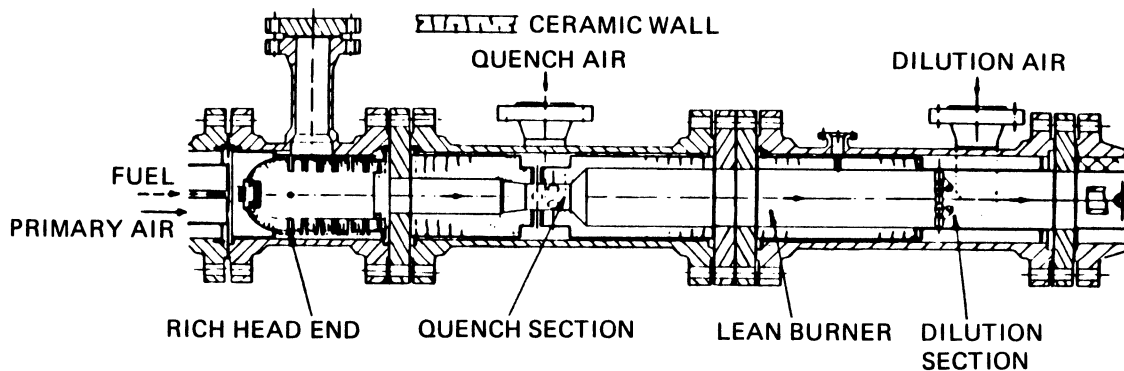
Figure 3-1.   Rich/Lean Configuration No. 1 Modified Westinghouse B-4
               Rich Burner, Venturi Jet Quench, and Lean Burner Pipe

BASIC DESIGN DATA

Configuration 1 was designed for the following nominal conditions:

Total air flow	1.81 kg/sec	-	4.0 lb/sec
Fuel flow	0.0358 kg/sec	-	0.079 lb/sec
Pressure	1.3 MPa	-	190.0 psia
Inlet temperature	370.0 °C	-	700.0 °F

RICH BURNER

The rich burner design was derived from the Westinghouse B-4 combustor. The major difference between the two is that the B-4 burner was constructed with conventional, cooled metal walls; the rich burner in this study has cast ceramic walls to accommodate higher gas temperatures (1750°C) associated with rich combustion. A ceramic wall was also selected because it does not involve extensive development. This particular ceramic was an alumina-based, phosphoric acid-bonded, ramming mixture, which was cast into suitable sized rings for insertion into the passage.

In the burner shown in Figure 3-2, the basic hot ceramic wall is inside a metal sleeve, which has an external air sleeve. Primary air for combustion cools the metal sleeve, then returns and passes through ports into the combustion region. The primary hole configuration, its total area, and the diameter and length of the section corresponds to the B-4 design. However, operation will be on the rich side (fuel/air equivalence ratio of 1.6), whereas the B-4 ran lean. Consequently, the section was modified by sleeving or plugging the holes to adjust the I.D. by the thickness of the ceramic, and to change the overall length. This flexibility was achieved by assembling the burner from separate but tightly-fitting, cast ceramic rings of the same inner diameter (.076M). Of the seven rows of holes (Figure 3-2), the two rows closest to the head end were used during the entire program. Performance of the first arrangement was found to be satisfactory.

The conventional air-assisted fuel injector consists of a hollow-cone simplex nozzle. The simplex nozzle was mounted into a mixing chamber where swirling air flow intercepted the fuel cone and reduced the cone angle to 25 degrees. The fuel exited from the mixing chamber through a square-edged orifice.

The nozzle assembly was surrounded by a cooling air shroud that was moved by the combustor pressure drop. This air provided coolant for the dome (the half-sphere closing the burner). Sweep air for the nozzle face and
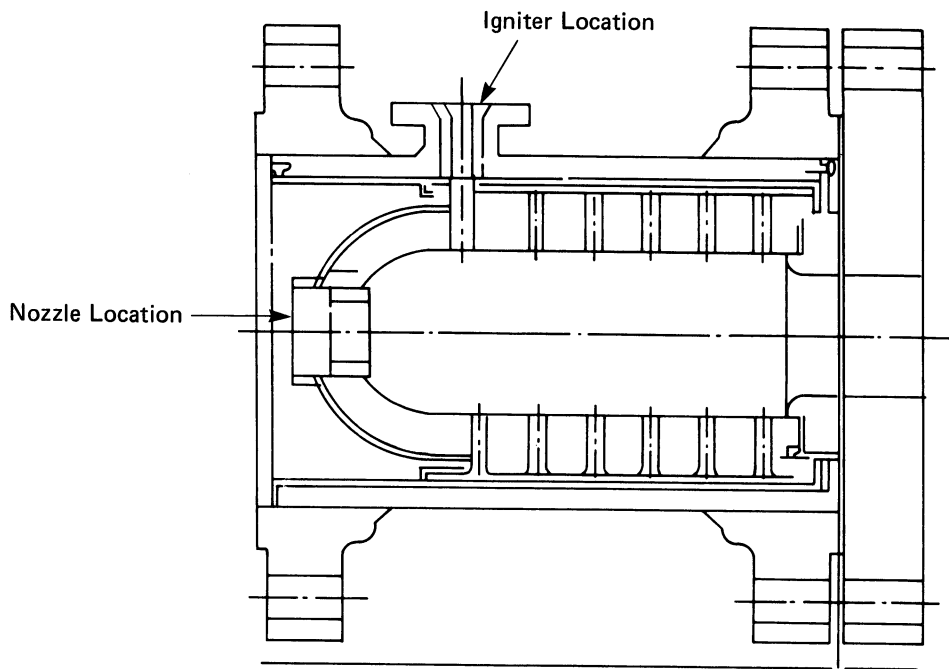
Figure 3-2. Modified Westinghouse Rich Burner With Ceramic Wall for
            Configurations 1 to 4

some of the air directly participated in primary combustion. The atomiz-
ing air was approximately 1 percent of the primary rich burner air flow;
9 percent of the primary air was the shroud air; and 90 percent entered
through the two rows of holes.

The nominal air flow in the rich burner was 0.3 Kg/sec (0.7 lb/sec). The
holes were sized to provide good mixing and to counteract the lack of
premix as much as possible. However, in sizing the holes it was intended
to maintain the hole pattern of the proven B-4 burner as much as possible.
The solution was to use two rows of .013M (0.53-inch) diameter holes,
with eight holes in each row. The other rows of holes (Figure 3-2) were
blocked.

QUENCH SECTION - VENTURI QUENCH

Quenching must be accomplished as fast as possible; otherwise, the oxygen content of the quench air will induce high temperature combustion reactions of the fuel in the rich burner exhaust, and thermal NO_x will be formed. Therefore, the quench air jets were closely connected with the rich burner exhaust and air was rapidly injected. This operation was achieved by reducing jet diameter and increasing jet velocities.

The number and size of quench zone holes have been determined from a jet penetration correlation. This correlation suggests that good mixing is achieved with four, .016M (5/8-inch) diameter holes with a momentum loss of 2 percent. The throat diameter is based on a 2.8 area ratio and becomes 0.06M (2.39 in.) for a 0.1M (4-inch) internal diameter of the hot wall combustor. For flexibility, eight holes were provided; plugging and unplugging is easily accomplished when necessary.

The nominal air flow in the quench section was .55 Kg/sec (1.2 lb/sec). As in the primary combustor, all approach, throat and diffuser walls are ceramic-(castable) lined.

LEAN BURNER

It is expected that the quench section will discharge a well mixed, high temperature (980°C-1100°C/1800°F-2000°F) reacting gas mixture; and given enough time, the combustion reactions will be completed in a straight pipe. In Configuration 1, there are no other means to bring about secondary combustion, but a second igniter that served as the lean burner was installed in the pipe entrance. During the program, there was no need to use the igniter. It was believed that the lean burner could be improved by installing a step to provide local (toroidal) recirculation. However, this adjustment was not necessary.

DILUTION SECTION

Dilution air was introduced through four holes and eight tubes, all 0.02M
(3/4 inches) in diameter. There were four radial tubes (.025M long), and
four .05M (2 inches) tubes at a 45-degree angle to the axial direction
pointing downstream. This arrangement was selected for mixing purposes.
Since the outlet temperature was measured at about .3M (12 inches) down-
stream of the dilution air inlet, it is possible that some of the thermo-
couples were influenced by cooler streamlines caused by the diluent air.

3.2   DIRECT INJECTION, VORTEX QUENCH - CONFIGURATION 2

Configuration 2 was designed as an alternative to the venturi quench
section of Configuration 1. Configuration 2 uses the vortex mixer
concept.

The planned vortex quench system is based on Westinghouse experience in
magnetohydrodynamics (MHD), which determined that combustor performance
depended on the effectiveness of vortex mixing. Vortex systems yield ex-
cellent fuel/air mixing, which results in complete, efficient combustion
reactions. The system shown in Figure 3-3 uses two vortex admissions;
the second counteracts the swirl from the first and produces axial flow.

The arrangement of vortex disks in the venturi throat is shown in Figure
3-4. As in the radial port design, it was desirable to create a flexible
disk arrangement. Any modifications can be accomplished by using various
numbers and combinations of disks in the throat. Since the total throat
length is 0.1M (4 inches), disks may be placed in series. These disks
can have their slots facing with supporting flow or counter flow. The
slots may be tangential, secantial or radial.

Quench Air

Rich End

Head End

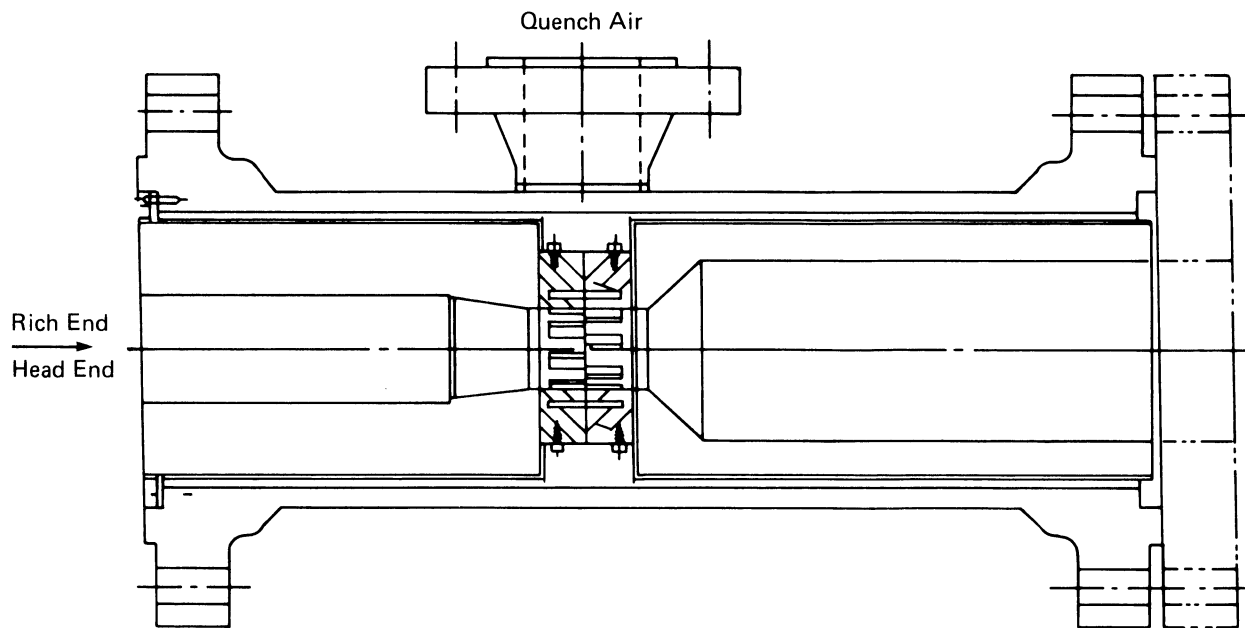Figure 3-3.   Opposing Swirl Quench Module of Configuration 2

$45° \pm 1°$ Typ

2.250
Dia.

For Tangential
Inflow Operation

$5.25 \pm .01$
Dia

For Secantial
Inflow Operation

44-025
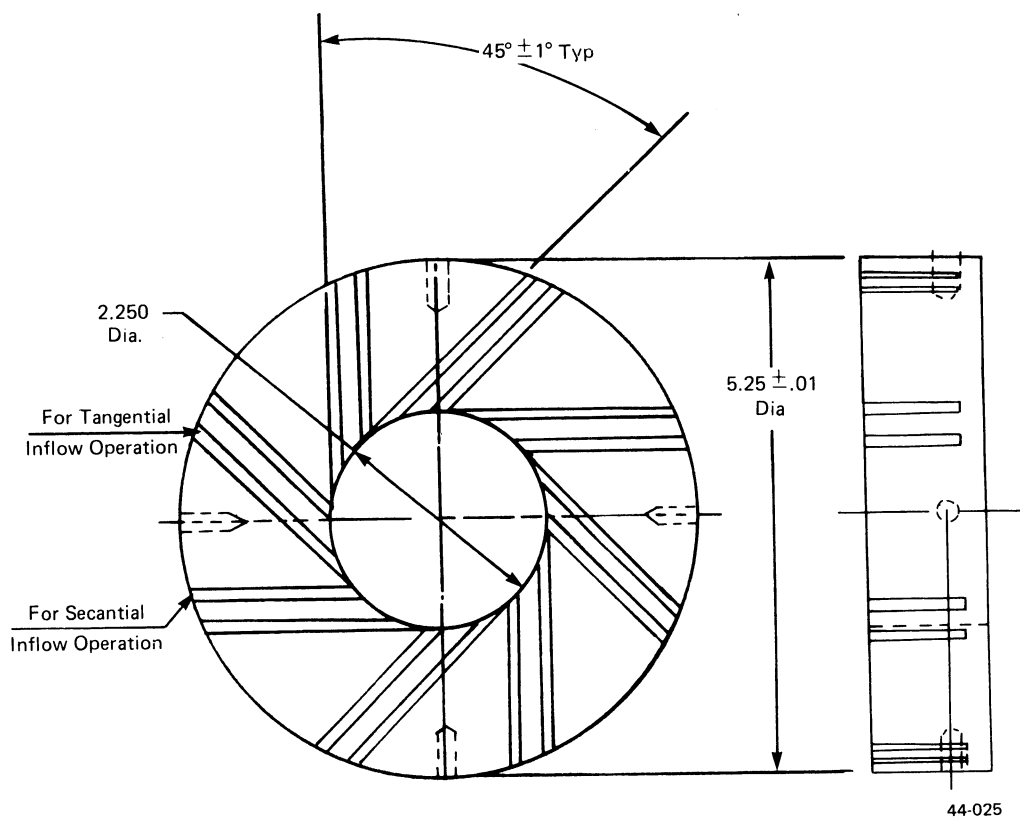
Figure 3-4.   Configuration 2 - Quench Section - Swirler Details

Eight slots were used in each case. One of the disks can be a blank, or
the disks can be mounted so that their solid portions are back to back.
Since tangential slots yield good mixing but poor penetration (a combina-
tion of radial and tangential components), a form of secantial slots were
selected for optimum mixing and penetration.

An inner diameter of 0.06M (2.5 inches) was selected to fit the Config-
uration 1 rich burner; the outer diameter has a plenum chamber for uni-
form inlet flow (Figure 3-3). Slot sizes were based on the assumption
that .55 Kg/s (1.2 lb/sec) of air at 370°C (700°F) and 1.2 MPa (170 psi)
will be injected (nominally) for a volumetric flow of .08M^3/sec.

3.3   DIRECT INJECTION, VORTEX QUENCH, CATALYST - CONFIGURATION 4

The approach to the design of Configuration 4 was very similar to that of
Configuration 1. Both consist of a rich burn module, a quench module,
and a lean burn module in series. The significant difference between
them is that Configuration 4 used a catalytic element as the lean burn
module because this element permits the lean burn module to better
approximate equilibrium operations with a commensurate reduction in
unburned hydrocarbons, CO and smoke.

Within the catalytic combustor element, homogeneous gas-phase reactions
occur simultaneously with catalytic reactions in the catalyst channels.
The homogeneous reactions initially occur close to the catalyst surface
since its temperature is very close to that of the adiabatic flame
temperature. As the bulk gas temperature continues to increase, the zone
of homogeneous burning spreads away from the catalyst wall to a larger
portion of the total gas stream. Eventually, the entire gas stream
reaches a temperature that supports homogeneous reaction. This process,
shown by Carrubba for the lean catalytic combustor (1, 2), is known as
CATATHERMAL* combustion (a catalytically-supported thermal combustion).

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* CATATHERMAL is a registered trademark of Engelhard Minerals and
  Chemicals Corporation.

In this process equilibrium is most closely approached while stable combustion and a uniform exit temperature are maintained. Design criteria for the catalyst is reviewed in Section 3.10.

DESIGN OF CONFIGURATION 4

The catalyst nominal design conditions are:

- Pressure   -   12 ATM
- Inlet Temperature   -   750-800°C
- Exit Temperature   -   1200°C
- Inlet Reference Velocity   -   25 M/S

Published data (3) was used with these operating conditions to design the catalyst tested in Configuration 4. A comprehensive set of data for two different catalysts was obtained from the DOE Low-Btu program (1). The most complete set of data included durability test results with exposure to H_2S. This data base was obtained with a Zirconia Spinel support, which was not considered viable for high temperature service. The proposed support, $\alpha-Al_2O_3$ Torvex, gave equivalent combustion performance. High temperature exposure data showed that high combustion efficiency occurred up to 1460°C. Performance was measured at the reference adiabatic flame temperature after each high temperature exposure (3). In order to ensure adequate turndown capability during the current test program, a .1M (4-inch) configuration was selected.

The assembly of Configuration 4 is shown in Figure 3-5. The catalytic element is in the lean burn module, and the rich burn and quench modules are the same as those in Configuration 2.

AIR   AIR   AIR

FUEL

RICH ZONE
CERAMIC LINED

QUENCH
ZONE

LEAN ZONE
CERAMIC LINED

CATALYST

DILUTION
ZONE

OPPOSING
SWIRL

CATALYST

A/C 9457

E.D.S.K. 355253
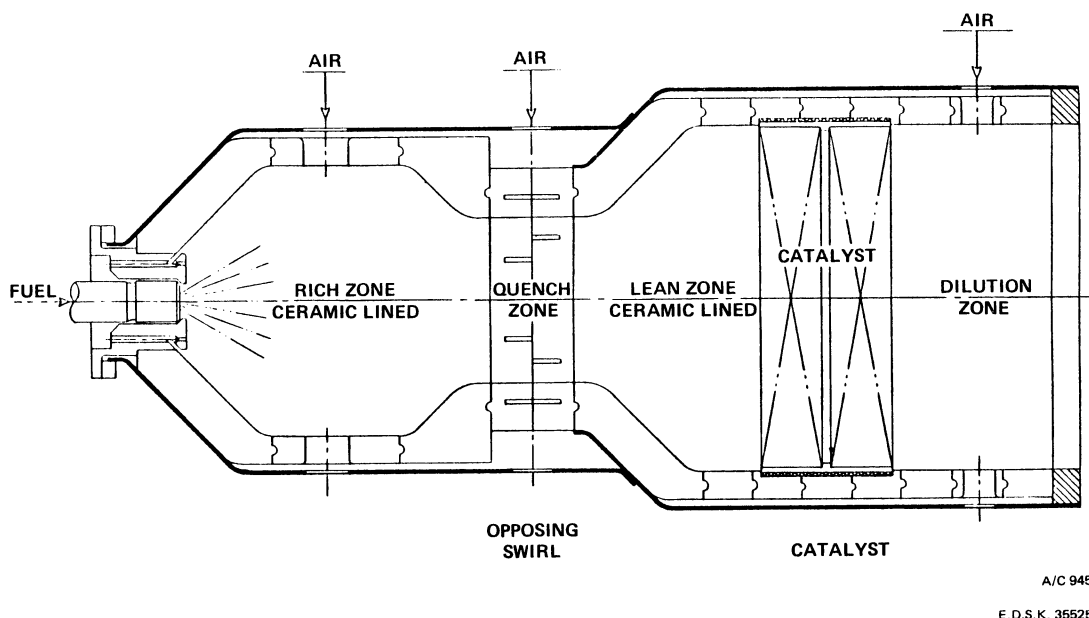
Figure 3-5. Staged Combustion - Lean Burn Catalyst Module

3.4   RECIRCULATING COUNTER SWIRL VENTURI QUENCH - CONFIGURATION 5

Configuration 5 is similar to the first four designs in its modular
arrangement - separate rich burner, quench section, lean burner and
dilution section. However, the rich burner module is actually a fuel
preparation module that supplies a premixed, prevaporized fuel-rich
mixture. Flame holding is accomplished through a recirculation pattern
created by concentric, counter-rotating, annular swirlers. This counter-
swirl flame holder supplements the pipe section with the B-4 burner in
it.

FUEL PREPARATION

Figure 3-6 shows the arrangement of the premixing/vaporizing module.
Primary combustion air (approximately 370°C/700°F) surrounds the injector
assembly. Since this version is designed to feed a 0.075M (3-inch)
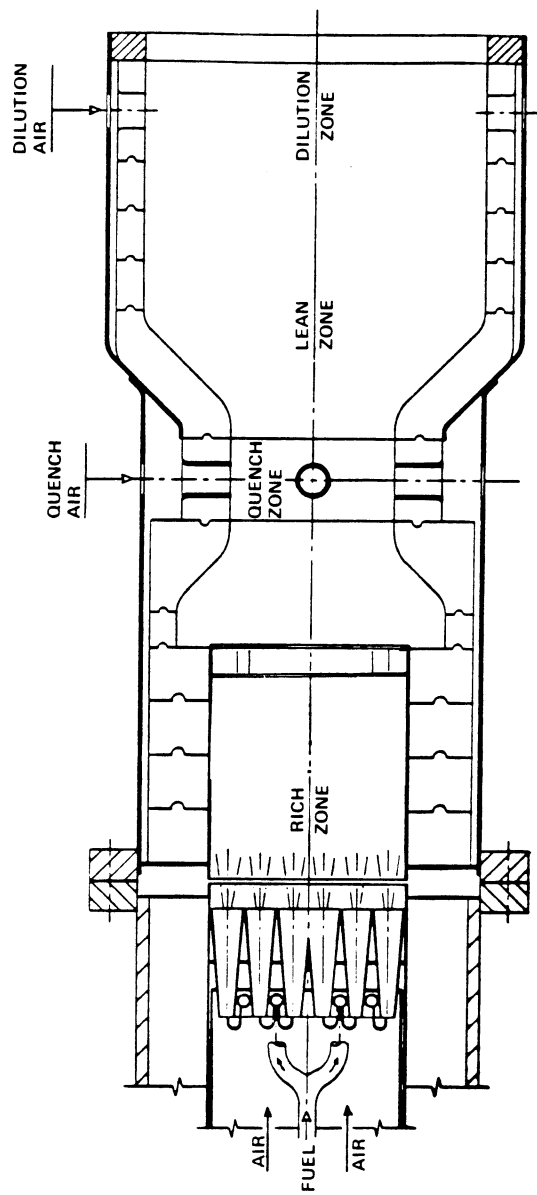
Figure 3-6. Recirculating Counter Swirl Venturi Quench - Configuration 5

diameter rich burner, the primary combustion air is carried in a 0.075M
(3-inch) diameter pipe. The cross section of this pipe is divided into
ten small passages consisting of slightly conical pipes. The fuel is
supplied through a conduit at the centerline of the pipe. The conduit
divides into two branches before reaching the injector assembly; by means
of an intricate manifold system, the two branches supply fuel to ten
hypodermic needle-type injectors in each of the ten air passages.

The counterswirl flame holder, which receives the fuel and primary air
mixture from the fuel preparation module, consists of turbine stator-like
annular vane rows. The pattern of rotation of each annuli is different.
The result is a flow pattern creating very strong shear regions that pre-
vent the formation of lean pockets. This complex swirl pattern collapses
downstream of the head end and provides a recirculation zone, which pro-
vides flame holding.

In the reference design, three concentric annuli create the shearing flow
pattern. The inner and the outer annulus give the flow tangential
velocity in the clockwise direction; the annulus between the two (annulus
"B") rotates the flow counterclockwise. This swirl arrangement was
derived from Configuration 9 (Multiannular Swirl Burner) where a free
vortex flow pattern was approximated. Startup is at the center, and the
deviation angle from the axial decreases in the outward direction. A
free vortex arrangement produces intensive shear layers between adjacent
annuli with different tangential velocities; the counterswirl increases
this effect.

As in the first four designs, the nominal air flow in this rich burner is
.33 Kg/s (0.72 lb/sec); the fuel flow is 0.3 Kg/s (0.07 lb/sec). This
air flow gives ϕ = 1.6 rich burner equivalence ratio through which
fuel-bound nitrogen is converted into compounds other than NO_x. The .03
Kg/s (0.08 lb/sec) fuel flow gives 1150°C (~2100°F) turbine inlet tempera-
ture at the nominal air flow rate of the facility - 1.9 Kg/s (4.2 lb/sec.).

3.5 HYBRID PILOTED RICH BURNER, VENTURI QUENCH - CONFIGURATION 6

In the original program work plan, it was anticipated that fuel-rich combustion, which prevents conversion of fuel bound nitrogen (FBN) into NO_x, requires an extensive reduction of lean pockets in the rich combustor. It was not believed that such a reduction was possible in a conventional combustor. Therefore, a premixed rich combustor was considered.

Given the advantages of the conventional, non-premixed fuel supply, as opposed to the premixed burner, Configuration 6 was devised as a compromise between the completely premixed and the conventional rich burner. It has the same downstream modules as Configuration 1: opposing jet venturi quench, straight pipe lean burner, and the standard dilution module. However, the rich burner consists of two combustors: a swirler-type rich burner in parallel with the B-4 burner (non-premixed) preceded by a fuel/air premixer. The rich burner is a hybrid burner that uses premixed or non-premixed fuel.

The sectional drawing in Figure 3-7 shows the rich burner of Configuration 6. The primary air is split in two flows, and most of it is directed into the "side arm" where most of the fuel is injected. There is 0.7M of straight pipe in which fuel mixes with the air. The active length of the tube can be changed by moving the location of the fuel nozzle. At the pipe exit, the fuel-rich mixture enters a centripetal swirler mounted coaxially with the B-4 burner; the hot exhaust of the B-4 burner ignites the mixture, and the two flows combine and proceed toward the quench section.

In order to prevent the fuel droplets from collecting on the wall of the mixing tube, a porous wall design was used. The air coming from the side arm enters the mixing tube through the cylindrical wall. As it emerges from the wall, it prevents the droplets from leaving the main air stream.
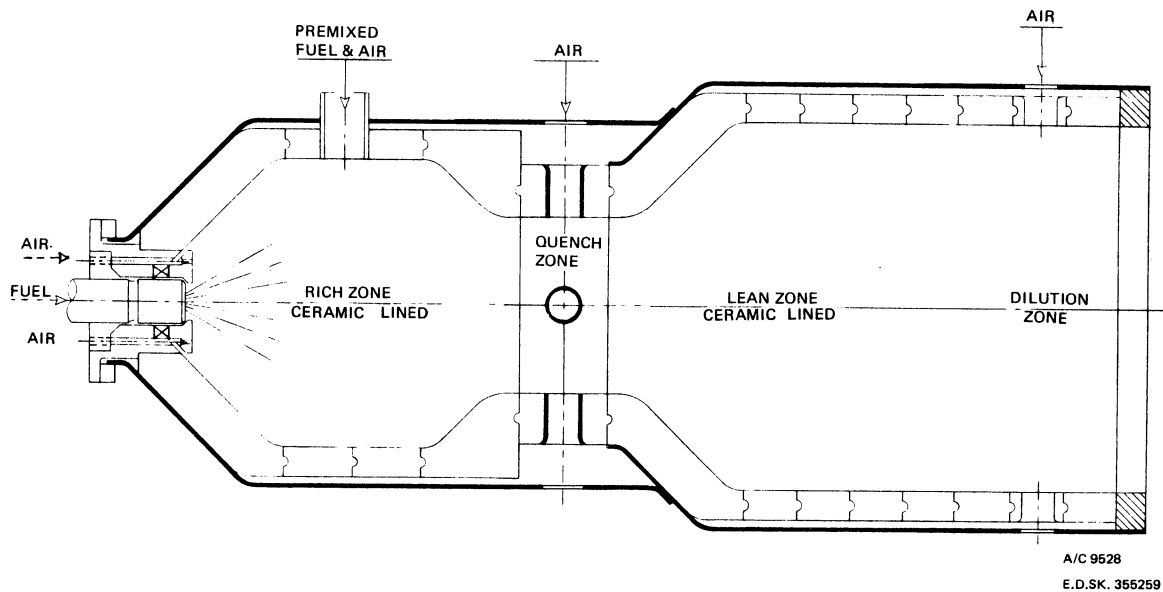
Figure 3-7.   Configuration 6 - Rich Hybrid Premix/Direct Injection

Since most of the air is premixed with the fuel in the side arm, even a lean B-4 burner (were this necessary for flame holding) cannot generated a substantial amount of NO_x from FBN.

In order to further improve the mixing performance of the B-4 burner, all the air is admitted through an axial swirler around the air-assist nozzle. The swirler is identical to the innermost annular swirler of the Multiannular Swirl Burner (MASB - Section 3.7, Configuration 9.) It fulfills the same purpose: the axial swirl collapses downstream of the fuel nozzle and creates a recirculation zone that serves as a flame holder. It is expected that the swirler-type flame holder is a better mixer than the conventional, opposed jet arrangement of the B-4 burner. Consequently, all radial holes of the B-4 are closed in this application.

The original objective in splitting the primary air flow was to divert 80 to 90 percent of the air into the side arm to reduce the B-4 burner to a flame holder. However, during the design phase, it was not known that the porous tube and the swirler had more flow resistance than the B-4 burner, and that a flow split from only 40 to 60 percent instead of 20 to 80 percent would be achieved. A smaller orifice with a very large pressure drop had to be installed upstream of the B-4 burner to achieve the 40-to-60 air split. The side arm orifice served only as a flow measuring device.

3.6 RICH CATALYST VENTURI QUENCH - CONFIGURATIONS 7 AND 8

The design of Configurations 7 and 8 was similar to Configuration 1. It consisted of a rich burn module, a quench module and a lean burn module in series. However, in Configurations 7 and 8 the use of a catalytic element in the rich burn module was evaluated for the following reasons:

- The rich catalytic element allows the rich burner to operate at higher equivalence ratios without soot formation since the catalytic element permits the rich burn module to better approximate equilibrium operations.

- The heat release rate of the rich module is appreciably higher than that in Configuration 1 because the reactions are completed within the catalyst monolith.

- The composition and temperature of the gas entering the quench module are uniform and predictable for this configuration.

In this configuration, the catalytic combustor element performs in the same manner as in Configuration 4 (Section 3.3).

CONFIGURATION 7 - CATALYST DESIGN

The following were the nominal design operating conditions of the catalyst:

- Pressure   -   12 ATM
- Inlet Temperature   -   360°C
- Exit Temperature   -   1200°C
- Inlet Reference Velocity   -   7 M/S
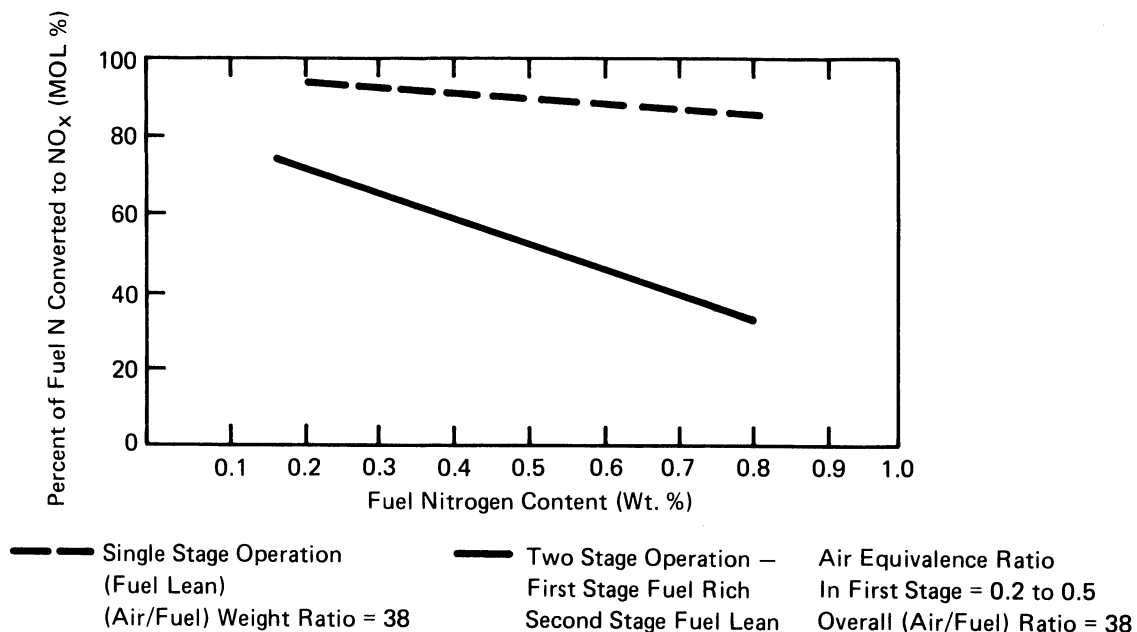
Figure 3-8. Comparison of Two Stage Vs. Single Stage Combustion of a Nitrogen-Containing Fuel in the Presence of a Catalyst

In addition to these operating conditions, published data (4,5) was used to design the catalyst tested in Configuration 7. Figure 3-8 shows an example of reduced FBN conversion to NO_x when operating a catalyst as rich/lean combustor on #2 fuel oil and propane.

The selected design is based on the Engelhard DXE-442 catalyst used in the NASA 5 ATM life test (6). The recommended support for the second segment is changed to 4 inches (.1M) of Torvex αAl_2O_3. This has the same geometric surface as 2 inches (0.5M) of zircon composite but provides twice as much residence time for homogeneous combustion reactions with less of a pressure loss.

CONFIGURATION 8 - CATALYST DESIGN

The following were the nominal design operating conditions of the catalyst:

- Pressure   -   12 ATM
- Inlet Temperature   -   315°C
- Exit Temperature   -   1425°C
- Inlet Reference Velocity -   8 M/S

Configuration 8 differs from Configuration 7 in that base metals have been added to the DXE-442 catalyst to provide a higher maximum operating temperature. Engelhard studies have shown that base metal oxides improve the durability of a catalyst system exposed to 1425°C (2600°F) adiabatic flame temperature operation. A limited (100-hour) durability test was run using #2 fuel oil. Test data was generated in six-hour segments. This data was obtained with the Engelhard high temperature reactor, which allowed "in-bed" thermocouple measurements to be made. This test demonstrated durable ignition capability with exposure to high fuel/air ratios.

Performance was also measured at an adiabatic flame temperature of 1260°C (2300°F) before and after the durability test. The results showed consistent performance at the design fuel/air ratio. Emissions performance was shown to be adequate.

FUEL INJECTION DESIGN

The fuel injector used in this configuration is based on the multiple conical tube injector developed by Tacina (7, 8). This injector is comprised of 10 discrete fuel injector tubes that adequately mix the fuel and air that enters the catalyst. This fuel preparation zone design was selected instead of other arrangements because of successful development and testing in related programs.

CONFIGURATION ASSEMBLY

The assembly of the fuel injector and the catalyst for Configurations 7 and 8 is shown in Figure 3-9. This assembly incorporates all the above design considerations.

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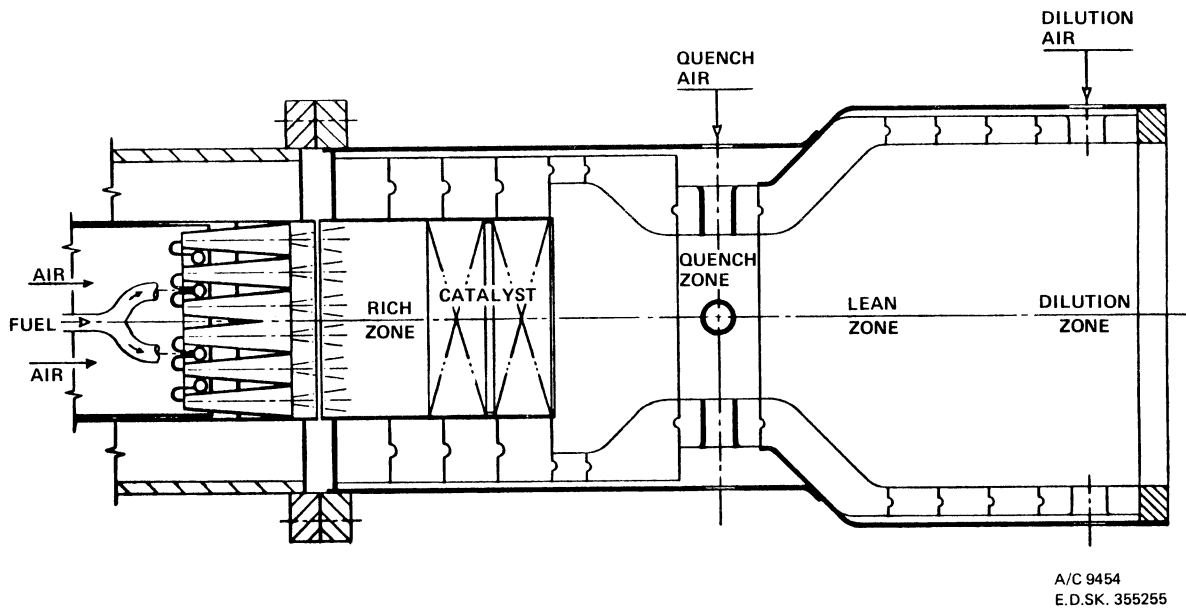
Figure 3-9. Catalytic Staged Combustion

3.7   MULTIANNULAR SWIRL BURNER - CONFIGURATION 9

The Multiannular Swirl Burner (MASB) is one of the two non-modular
burners used in the program. The thermochemical process in the MASB is
the same as that of the modular combustor: rich burn, quench, lean burn
and dilution. However, the MASB performs these functions in one or two
instead of four modules. The functions of the rich/lean burner are
performed in an axisymmetric arrangement in which the combustion chamber
is subdivided into concentric annuli as shown in Figure 3-10.

Based on Beér's design*, the burner consists of a system of axially dis-
placed concentric rings. Each pair of radially adjacent rings forms an
annulus. The axial displacement of the annular rings is such that the
envelope of the ring tips form a divergent passage. This arrangement
permits tight control of the axial and swirl velocity distribution in the
burner.

Because of the MASB arrangement, all of the combustion air (or fuel/air
mixture) axially enters each annulus. The desired tangential velocity
distribution is produced by the flow passing through blade rows in the
individual annuli. The controlled swirl velocity distribution results in
a strong radial transfer of the angular momentum in the flow. This
results in high turbulence. There is extensive mixing between the
adjacent annular flows due to the formation of concentric layers in which
the turbulent shear stresses are maintained at a high level.

Another feature of the burner is that, as a result of the combination of
swirling flow and the divergent passage formed by the concentric nozzles,
a toroidal recirculation zone is formed in the central region of the
burner. The size and strength of this recirculation zone is dependent,
for a given divergent passage geometry, upon the axial and swirl velocity
distribution in the burner.

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* British Patent 45652 (1965)
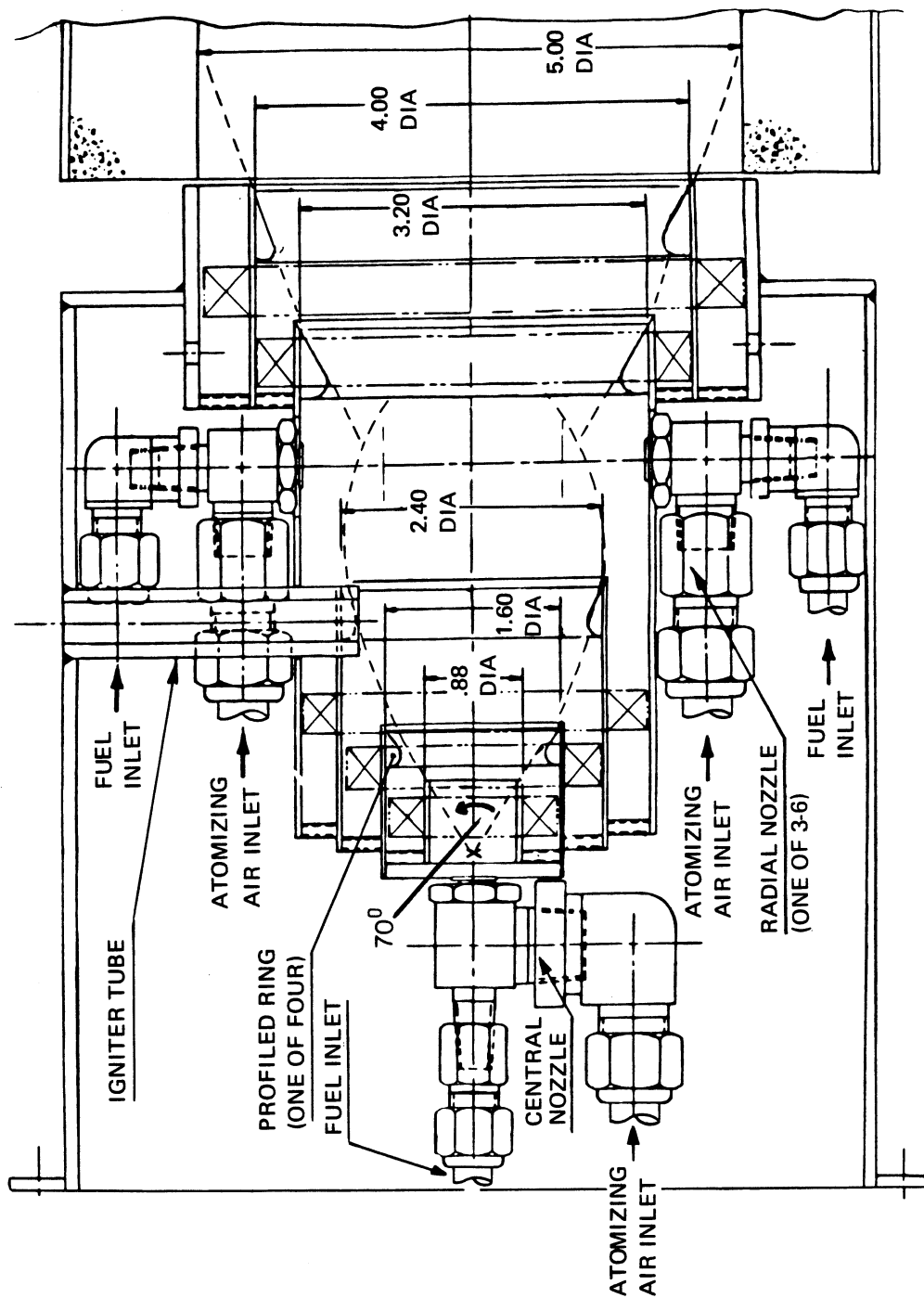
MULTIANNULAR SWIRL BURNER (MASB)

Figure 3-10. Multiannular Swirl Burner (MASB)

GV 9853

When operated in the staged combustion mode as a low NO_x burner, the central nozzle runs fuel-rich, and the fuel-rich zone forms a partially-stirred recirculation zone in front of the central nozzle. The fuel-rich fuel gases are mixed with cold air entering through radially and axially displaced annuli along the boundaries of the recirculation zone. The air rapidly reduces the mixture temperature below a limit conducive to thermal NO_x formation. Following this rapid "quench," the air from the outer annuli is introduced and the lean combustion stage begins. Since the burner is cooled by the combustion air flowing through the annular rings, there is no need for film cooling air. Consequently, all combustion air is available to regulate the flow and mixing patterns in the burner. Here, combustion is completed in a temperature range that is high enough to burn out soot and combustible gases, but low enough to minimize the formation of thermal NO_x.

The hot recirculating gases in the central, toroidal vortex also provide flame holding, and permit stable burner operation over a wide operating range without the need for bluff body stabilizers or refractory surfaces directly exposed to the flame.

Fuel with chemically-bound nitrogen was expected to burn in this swirl combustor with low emissions of NO_x. Axial staging of fuel and/or air can be easily accomplished with this design so that the rich burner section may be used to reduce NO_x. Another advantage of the MASB is that the turbulence is uniformly high throughout the combustion volume. This condition, in addition to strong back mixing, limits carbon formation because the onset of smoke formation shifts to a lower oxygen/carbon atomic ratio of a premixed hydrocarbon/air mixture.

Fuel is injected through the central nozzle (Figure 3-10) and through three nozzles into the third annulus. Air atomizing nozzles are used to achieve the desired drop size (20 micron).

OPERATIONAL MODES OF THE BURNER - FLEXIBILITY

This design is flexible in terms of the fuel and air supply arrangement.
In order to change the equivalence ratio of the rich burner (the primary
zone of the burner) for a given total fuel flow, the MASB was used in
conjunction with the dilution module of the modular configuration. Since
the dilution section has a separate, metered air supply, it is possible
to shift air away from the burner into the downstream dilution section.
(The nominal air flow of the burner was 3 Kg/s (6.5 lb/sec); the air
shifted into the dilution section was between 0.2 and 0.8 Kg/s). Because
the exhaust temperature of the burner (the temperature downstream of the
largest swirler in Figure 3-10) is always below thermal NO_x-producing
temperature levels, shifting the air from the reaction zones to the
dilution section affects NO_x formation only in the head end. This result
was the objective of the investigation.

The MASB can be designed with or without a separate dilution section once
the desired flow split between dilution air and reaction air is established.
An increase or decrease in the radial width of the last (fifth) annulus
is used to achieve the desired dilution flow. If a separate dilution
section is desired, the design tested in this program may be used. Test
results include exhaust temperature profile control and smoke burnout at
the exhaust temperature of the lean burner (rather than at the lower
temperature after dilution).

This design is also flexible because fuel flow as well as air flow can be
regulated. The head end equivalence ratio can also be changed by shifting
some of the fuel from the central into the radial nozzles.

The swirl distribution selected for this program was a free vortex design
where the annuli provided 60°, 50°, 40°, 30° and 20° flow angles in
relation to the axial direction. The largest angle was in the center
swirler (60°). This design was selected because of its low pressure drop
and high turbulence level. To turn the flow in the individual swirler
rings, thin circular arc vanes approximately 0.01M long were used. The
setting of the vanes to the air flow from the axial direction into the

desired angles of the various annuli was determined on the basis of NASA Report SP-36 (revised) (Aerodynamic Design of Axial Compressors, 1965).

The axial distribution (distancing) of the swirlers was arranged so that the succession of four, profiled rings carrying the swirlers formed the assumed contour of the flame. The following assumptions were used:

- The expanding, swirling flame near its inception (the innermost swirler) has a 35-degree half-cone angle. This angle and the central, blank cylinder (0.022M in diameter) define the location of the flame cone apex.

- By the time enough air enters the burner from the subsequent annuli to cause $\phi p = 1.6$ in the head end of the burner, the flame contour should be axial. This occurred in the third annulus; therefore, a smooth curve (circle) was drawn between the two tangents (the 35-degree tangent in the innermost swirler and the axial tangent of the flame contour in the third swirler); the profiled rings were fitted accordingly.

- The recirculation zone was a hemisphere downstream of the third annulus.

- A secondary combustion zone starts in the fourth swirler, again at 35-degree half angle; this zone ends with an axial flame contour in the fifth annulus.

- The outermost (maximum) diameter of the burner is based on an assumed 30M/sec reference velocity.

- The radial distribution of the annulus area was based on uniform axial velocity (free vortex criterion) and five radial annuli.

- The secondary fuel supply (radial nozzles) was installed in the location of the downstream end of the rich zone recirculation.

These key assumptions about the flow design were gradually developed in consultation with the inventor, Professor Beér, based on his experience with the atmospheric burner.

FLOW SPLIT

The rich/lean combustor is a staged combustion device although there is only one point of fuel introduction. A single fuel supply is a desirable

combustor feature, and it was planned to test the MASB first with only the central nozzle.

The innermost axial location of the single nozzle was at 'A' (Figure 3-10) with a possibility of moving it upstream. Through this arrangement, the spray direction can be changed in case of fuel overpenetration. (Overpenetration may be caused by an overly powerful atomizing air jet when very fine atomization is attempted.) The position of the central nozzle created an air gap between the nozzle and the 0.88-inch (0.22M) diameter tube. This caused ignition difficulties during the atmospheric test of the burner. By running the burner with the nozzle face at the apex of the presumed flame cone, the gap between the nozzle and the central tube was closed.

The position of the nozzle was not changed during the tests. For the fuel spray to clear the central tube, 50-degree cone angle (nominal) nozzles were used. In case the secondary combustion zone had to be richened, three radial nozzles were also installed. These were similar to the central nozzle, except that the fuel spray cone angle was 70 degrees (nominal): the larger the cone angle, the sooner the fuel is distributed on the perimeter.

The nominal capacity of the total fuel system was .22 M^3/h (60 gph). The entire fuel flow can be delivered through the central nozzle, or any portion of the total fuel can be directed into the radial nozzles.

Ignition was provided through the igniter tube using the same igniter as the other configurations in the program. A tube is not shown in the same radial flame as the igniter (only 90 degrees offset) through which a loss of flame was detected when running; ignition was confirmed at startup.

3.8   PERFORATED PLATE, VENTURI QUENCH - CONFIGURATION 10

The design of Configuration 10 is similar to that of Configuration 1; it consisted of a rich burn module, a quench module and a lean burn module in series. The signficiant difference is that Configuration 10 has a perforated plate acting as the rich burn module premixed, prevaporized fuel preparation zone.

The following were the nominal design conditions of the perforated plate:

- Pressure   -   12 ATM
- Inlet Temperature   -   315°C
- Exit Temperature   -   1760°C
- Inlet Reference Velocity -   8 M/S

Data (9, 10) derived from experiments with a lean burn perforated plate was used with these operating conditions to determine the design of the perforated plates used in Configuration 10. The perforated plate was machined from .014M steel with 37 holes on a triangular pitch. The hole diameters were either .006M or .004M for area blockage ratios of 74 or 86 percent, respectively.

Two fuel preparation zones were designed for Configuration 10. It was recognized that the reference velocity entering the perforated plate/rich burner would be very low; therefore, the first design used a static mixer for flow blockage in the fuel preparation zone to minimize the tendency for flashback. In front of this static mixer there was air-assist nozzle similar to the one in Configuration 1.

The other fuel injector used in this Configuration was based on the multiple conical tube injector developed by Tacina (7, 8). This injector was comprised of 10 discrete fuel injector tubes that adequately mix the fuel and air that enters the perforated plate.

3.9   ROLLS-ROYCE COMBUSTOR - CONFIGURATION 11

Configuration 11, the Rolls-Royce design shown in Figure 3-11, was based on scaling and modification of a production standard burner. When utilizing the rich/lean technique, this unit produces low levels of thermal NO_x. This burner has three zones, each controlling a phase of the total combustion process:

- Primary Zone
- Secondary Zone
- Quench Zone

In the fuel-rich primary zone, vigorous recirculation is generated by a swirler cuff and high penetration, low blockage secondary air ports. Fuel preparation, ignition and initial rich burn flame propagation occur in this zone. It was designed to operate with a fuel/air equivalence ratio of ϕ_F = 1.3 to 1.4. Because of the mixing process and the resulting excellent fuel/air preparation, it is possible to run richer than stoichiometric conditions without excessive smoke. Calculation of the primary equivalence ratio is based on the known fuel flow and a prediction of the air flow distribution within the zone:

$$\phi_F \text{ Primary Zone } = (F/A)/(F/A)_{stoi} = 1.3 - 1.4.$$

$$F/A = \frac{\text{Design fuel flow lb/s}}{\text{Primary zone recirculation airflow}}$$

Primary zone recirculation air flow = mSw + 1/3 mc + 1/2 msp.

where   mSw = Air mass flow through swirler

mc = Air mass flow through cooling upstream of secondary ports

msp = Air mass flow through secondary ports

The secondary zone operates at a design ϕ_F = 0.65, which ensures the completion of the reaction at a low temperature. This process produces a
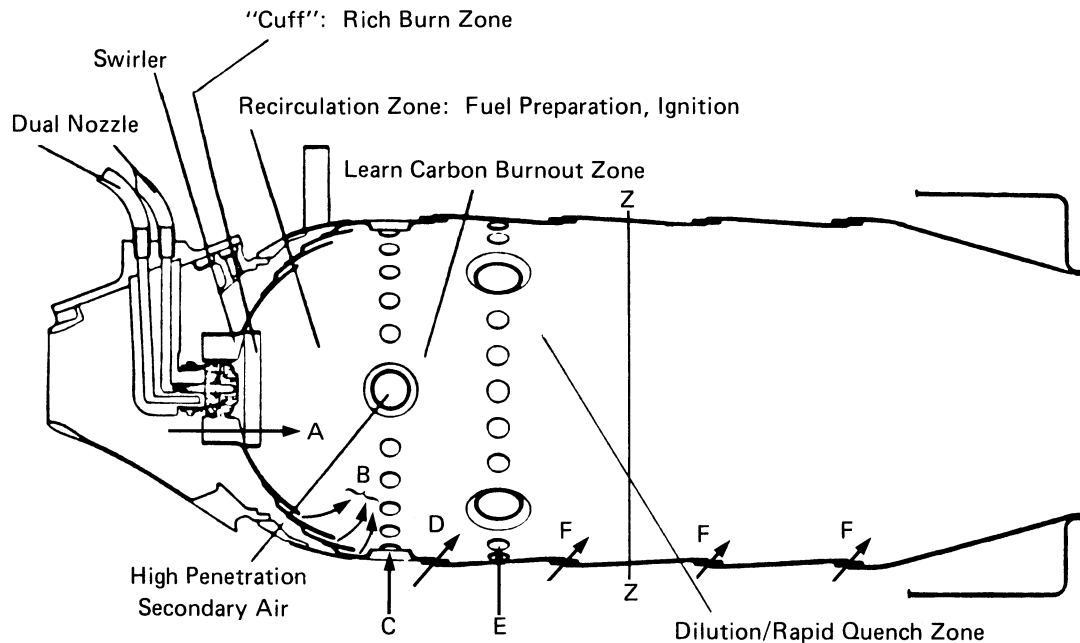
Figure 3-11. Rolls-Royce Combustor

minimum of thermal NO_x and permits consumption of any primary zone-generated carbon and unburnt hydrocarbons. Residence time in this zone is critical. It is a compromise between NO_x production and hydrocarbon burnout. Therefore, axial positioning of the quench zone is of major importance.

$$\phi_F \text{ Secondary zone} = \frac{\text{Design Fuel Flow}}{\text{Secondary air flow}}$$

where

Secondary zone air flow = mSw + mc + msp

Rapid quench is achieved by introducing quench air downstream of the secondary reaction. Four, high penetration jets ensure that quench air reaches the center of the combustor where the reaction occurs. In the Rolls-Royce design the air distribution is such that the rapid quench zone becomes the dilution zone.

3-28

Fuel injection is accomplished with a duplex pressure atomizing nozzle. The center primary nozzle, with a low flow number, is used for ignition; the outer main nozzle takes over shortly after ignition for running to design conditions.

Initial testing showed excellent combustor performance and thermal NO_x reduction, but fuel-bound nitrogen conversion was not optimized. To make the primary zone recirculation run richer (ϕ_F = 1.6 at design condition), a short test incorporating a swirler blank and minor blanking of the secondary ports was conducted.

3.10   LEAN CATALYTIC - CONFIGURATION 12

The design of Configuration 12 included a lean catalytic combustor using premixed fuel and air that will generate a minimum of NO_x. It also takes advantage of the catalyst's ability to burn lean mixtures at low temperatures.

The reason for the inherently low NO_x emission is shown in the hypothetical curves of flame temperature versus combustor length shown in Figure 3-12. In both the conventional diffusion flame combustor and the lean premix combustor, there is a high flame temperature region near the head end of the combustor. However, in the catalytic combustor, chemical reaction may occur without an open flame; therefore, the high temperatures conducive to NO_x formation are avoided. The catalytic element, which is a honeycomb device with a catalyst coating on a ceramic substrate, also acts as a flame stabilizer. It ensures steady, continuous combustion at the desired fuel-lean conditions. As a result, all fuel is burned and no slugs of a partially reacted mixture pass to the exhaust. There is very little carbon soot or unburned hydrocarbon in the exhaust. Combustion occurs in the catalytic element as it does in Configuration 4.
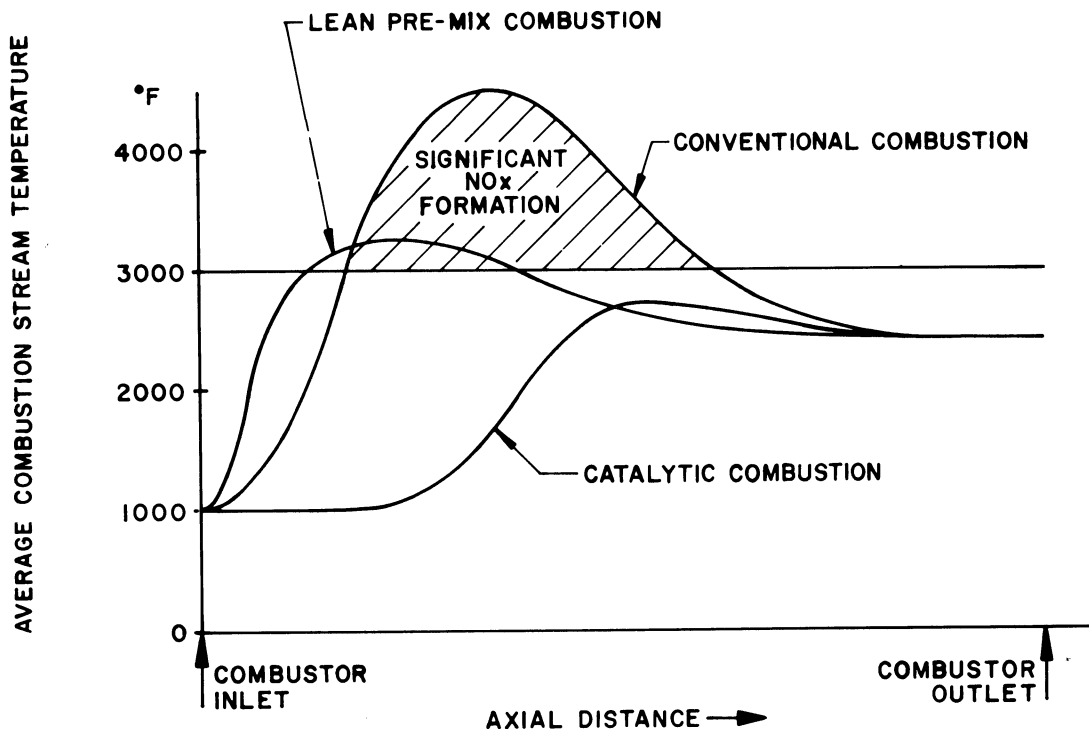
Figure 3-12. CATATHERMAL vs. Lean Premix and Conventional Combustion

There have been enough lean catalytic combustor experiments to generate
pertinent design information. The design criteria for this test are
summarized in the following subsections.

DESIGN INFORMATION - CATALYST

Minimum Inlet Temperature

As the description of the CATATHERMAL effect implies, a minimum tempera-
ture of the reactants is required to ignite the homogeneous reaction.
This temperature is not fixed, but is a function of the catalytic material,
its age, manufacturing technique, substrate geometry and fuel/air ratio.
Metal catalysts have a lower minimum inlet temperature than higher
temperature catalysts.

3-30

Minimum Fuel/Air Ratio

This is sometimes referred to as the ratio required for the minimum required adiabatic reaction temperature. Below a certain fuel/air ratio, the reaction will not be complete although the inlet temperature seems high. NASA has evaluated numerous catalysts from four manufacturers. Bulzan correlated the results (11) and found that for propane and test inlet conditions of 530°C and 3 atms, the minimum fuel/air ratio was correlated by the expression:

$$f/a \ (min) = (V_{ref}/L/K)^{0.1}$$

where f/a (min) is the minimum fuel/air ratio; K is the catalyst constant, which includes the effect of catalytic material, its age, catalytic loading, manufacturing technique and substrate geometry. A large value of K is associated with an efficient catalyst; V_{ref} is the reference velocity for the catalyst face area; and L is the catalyst length.

There are various methods to reduce the minimum fuel/air ratio in order to operate the turbine at a lower continuous load. However, as discussed below, all the known methods lead to higher pressure drops in the catalyst.

Maximum Reference Velocity

This is one of the most important parameters. If the velocity is too high, the reaction will not be complete. If it is too low, there is a long residence time in the fuel preparation zone that causes autoignition problems.

Maximum Operating Temperature

There is a maximum temperature at which a catalyst may operate without deteriorating. A typical maximum temperature for metal catalysts, such as the one used in Configuration 12, is 1260 to 1320°C. A typical temperature for advanced catalytic materials might exceed 1700°C (12).

Fuel/Air Ratio Distributions Entering the Catalyst

The fuel/air ratio distributions entering the catalyst are set by the maximum and minimum operating fuel/air ratios for the catalyst. The maximum allowable fuel/air ratio variation from the mean (which relates to the maximum operating temperature) is based upon any fuel/air ratio that has an adiabatic reaction temperature that exceeds the maximum operating temperature.

There is a restriction on the fuel/air ratio distribution for the minimum fuel/air ratio similar to that set by the maximum operating temperature. Even though a bulk fuel/air ratio is above the minimum fuel/air ratio, there will be local concentrations below the average. If these local concentrations are below the minimum, combustion will be incomplete.

Figure 3-13 shows one predicted acceptable operating range for fuel concentration distribution with an inlet temperature of 450°C. The important characteristics of the range are not the absolute values. Whatever the acceptable limits are, if the mixture in the fuel preparation zone is more homogeneous, then the combustor can operate over a wider range of fuel/air ratios.

Reactor Pressure Drop

The catalyst will operate with a pressure drop because of frictional losses in the small channels and momentum losses from heat addition. Supporting data in Figure 3-14 shows that the pressure drop increases with higher reference velocities and higher heat. Bulzan's correlation expresses the pressure drop at a constant reaction temperature of 1175°C as:

$$\frac{\Delta P}{P} = AKL \ (V_{ref})^{1.5}/P/F^{2}$$

where $\frac{\Delta P}{P}$ is the percentage pressure drop,

LIMIT (% OF MEAN)

MAXIMUM <u>POSITIVE</u> FOR LOCAL OUTLET TEMPERATURE = 2400F

MAXIMUM <u>NEGATIVE</u> FOR LOCAL OUTLET TEMP. = 1860F

ACCEPTABLE OPERATING RANGE

(FUEL/AIR) MEAN
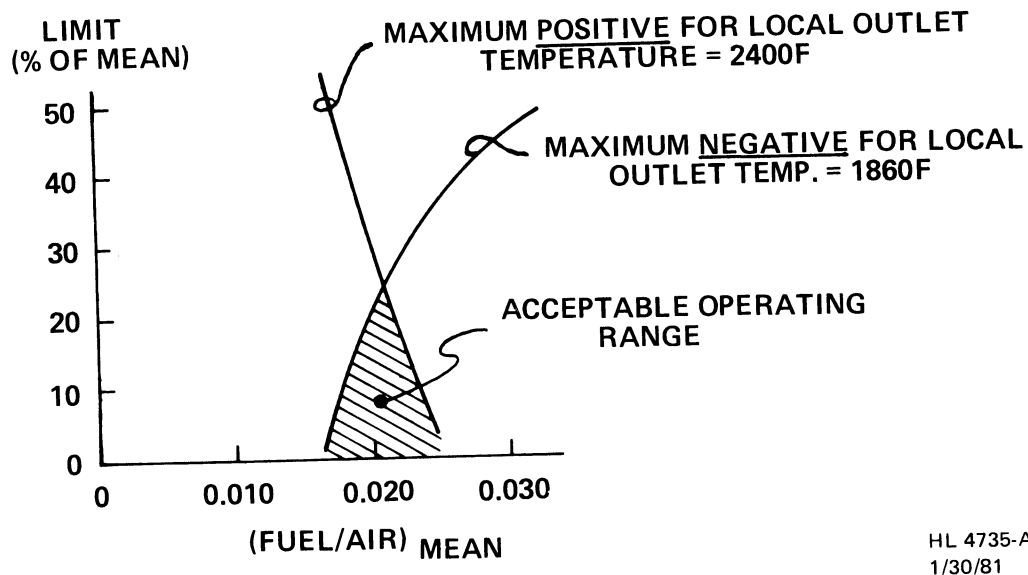
HL 4735-A
1/30/81

Figure 3-13.   Limits of Fuel Concentration Distribution Entering a
Catalyst (Combined)

A is a constant, P is the pressure, and F is the fractional
open area

A comparison of this equation with the minimum fuel/air ratio equation
indicates that factors improving the minimum fuel/air ratio do so to a
much lesser degree than the same factors that deteriorate the pressure
drop. Catalyst manufacturers have had difficulty in reducing the minimum
fuel/air ratio without sacrificing pressure drop. The data for both
correlations was obtained by testing metal as well as ceramic substrates.

Catalyst Aging

The effects of aging are not yet well defined. There is some deteriora-
tion to a stable level after initial firing; this was the level used for
the design. It appears that 1,000 hours of operation with clean oil will
not be a problem in a continuous load application (13).

3-33
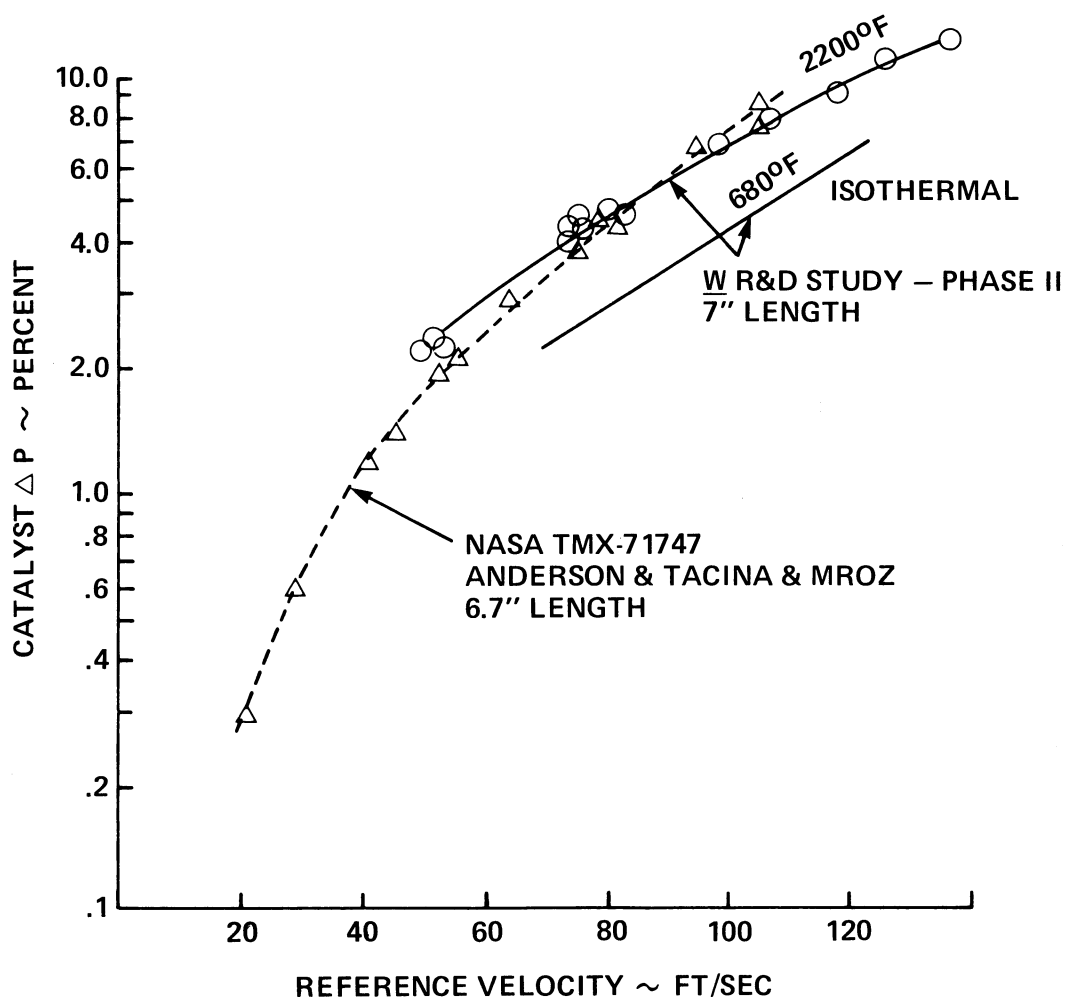
Figure 3-14. Catalyst Δp Vs. Reference Velocity

The effects of fuel contaminants on catalyst life are less well defined. Halogens are reported to be harmful, and sodium and potassium have some negative effect, but low boiling contaminants such as lead are not problematic.

The catalyst can mechanically deteriorate from cyclic thermal stresses and mechanical vibration.

DESIGN INFORMATION - FUEL PREPARATION

Fuel Vaporization

The fuel should be vaporized to an exact level in the fuel preparation zone. The actual degree of vaporization required for catalytic reaction is not known, but it appears to be related to the mixing requirement. The droplets, which can burn with a diffusion flame in the CATATHERMAL reaction zone of the monolith, must be quenched quickly by dilution inside the passageways so that they have a negligible effect on catalyst durability and NO_x generation. In addition, sufficient fuel must be vaporized so that the vaporized fuel/air ratio entering the catalyst is greater than the minimum fuel/air ratio. If the droplets are above a certain size, they will pass through the catalyst and produce afterburning (streak).

Fuel Autoignition

Autoignition is a self-induced, time-dependent combustible mixture reaction. Autoignition of the fuel/air mixture upstream of the catalyst is undesirable because it might overheat the metal basket, which results in increased velocity and pressure loss in the catalyst. Neither the temperature nor the increased velocity will destroy the catalyst because any temperature rise resulting from autoignition will remain below the maximum catalyst operating temperature in a well mixed fuel-preparation. This statement, however, must be qualified.

- In fuel preparation zones with multiple fuel-injection locations, the resulting diffusion flame sufficiently changes the aerodynamics; this results in a poorly mixed flow with extreme hot spots that can destroy the catalyst.

- Autoignition is potentially damaging to catalytic activity at the monolith entrance. The maximum catalyst operating temperature refers to the catalyst exit temperature where homogeneous reactions dominate. Some loss in catalytic activity is expected if the same temperature occurs at the entrance. To determine the significance of a loss in activity, tests must be run on a specific catalyst.

Autoignition will not occur if the fuel/air ratio is below the lean flammability limit. When the combustion zone has a mean fuel/air ratio below the flammability limit, there cannot be a condition with a locally flammable mixture after any residence time greater than the ignition delay time. Despite the effects of autoignition, there is still a thermal shock when the preheat flame is ignited. This could damage the catalyst.

Flashback into the Fuel Preparation Zone

The results of flashback are identical to those of autoignition in that gases with a similar temperature enter the catalyst. The phenomenon that causes flashback is different, however, because the upstream gases are ignited either by hot gases recirculating from the hot catalyst or by flames propagating upstream faster than the gas velocity downstream. Literature on premixing indicates that if the bulk velocity is greater than about 15 M/S, a flow cannot be created upstream. In certain modes if there is a pressure pulse, the flow can reverse for an instant and permit upstream propagation. By designing the unit without recirculation zones near the catalyst, the first cause of flashback can be prevented, but the second will be more difficult to prevent or predict.

Load Range Operation

There are several difficulties when the catalyst is required to operate in a normal load range. The most significant (Figure 3-13) is its narrow operating range. Innovative design and control methods are needed to operate a catalytic combustor from ignition to full load.

DESIGN OF CONFIGURATION 12

Catalyst Design

The following were the nominal design conditions of the catalyst:

- Pressure   -   12 ATM
- Inlet Temperature   -   360°C
- Exit Temperature   -   1200°C
- Reference Velocity   -   24 M/S

Published data (14, 15, 16) as well as Engelhard configuration develop-
ment data were used with these operating conditions to determine the
designs of the catalyst tested in Configuration 12.

Performance data was obtained by using three catalyst configurations.
They are multi-cell configurations, mirror image configurations and a
single-cell configuration. Pressure losses are significantly lower with
the mutli-cell configuration. It is designed to achieve a CATATHERMAL
operating temperature at the end of the first segment. Because this
design reduces pressure loss, it was selected for further testing. The
catalyst is an Engelhard DXE-442 catalyst (Pd/stabilized alumina) on two
inches of 256 channels per square inch Zircon composite, followed by four
additional inches of DXE-442 catalyst on 60 channels per square inch
Torvex α-Al$_2$O$_3$ (total length is six inches).

Fuel Injection Design

Figure 3-15 shows the fuel injector used in this configuration, which was
based on the multiple conical tube injector developed by Tacina. This
injector is comprised of 24 discrete fuel injector tubes, which provided
enough mixing of the fuel and air entering the catalyst. This fuel
preparation zone design was selected on the basis of successful develop-
ment and testing in related Westinghouse programs.
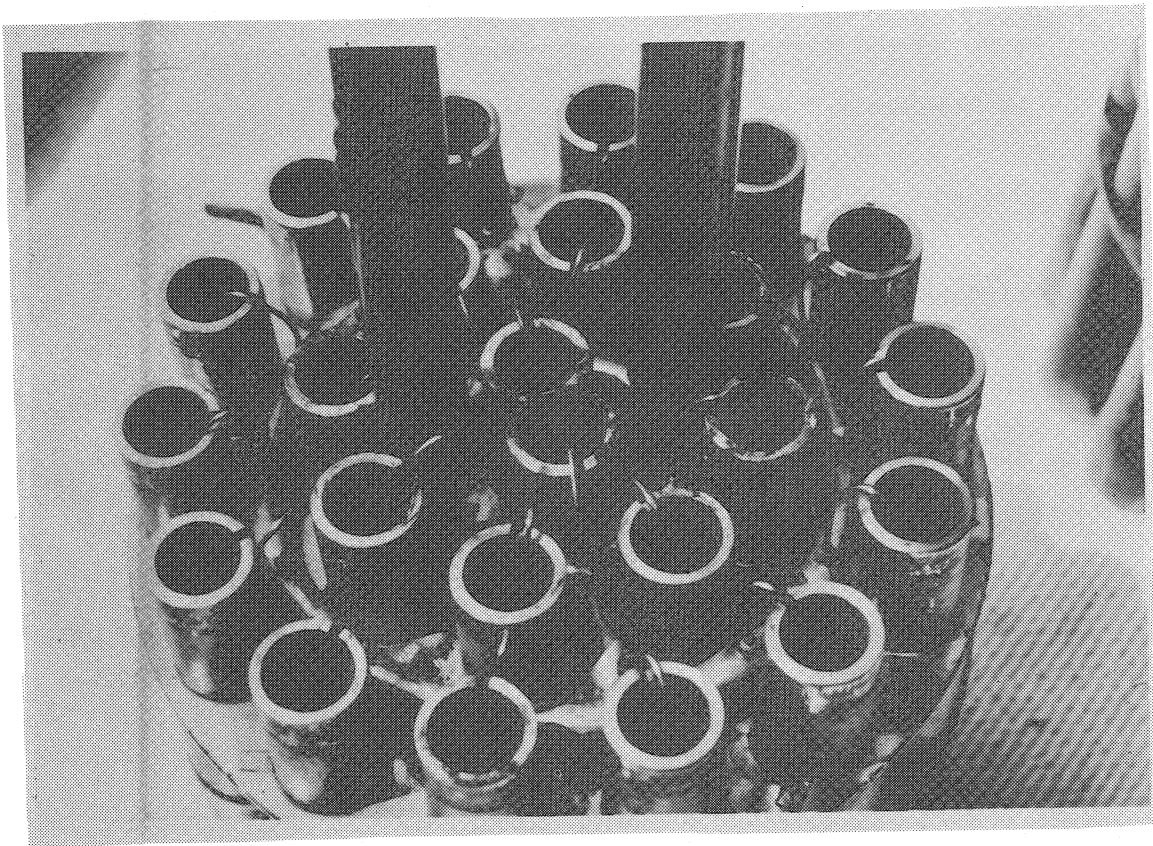
Figure 3-15. Fuel Injector for Prevaporized and Premixed Configurations

The assembly of the fuel injector and the catalyst for Configuration 12 is shown in Figure 3-16. This assembly incorporates all the design considerations previously discussed.

3.11   LEAN HYBRID PREMIX/DIRECT INJECTION - CONFIGURATION 13

The hardware of Configuration 13 is identical to that of Configuration 6. The concept for this configuration originated from the concern that a non-premixed burner has pockets of fuel/air mixtures, which significantly differ from the calculated average fuel/air ratio. With a lean burner, this difference means close-to-stoichiometric combustion for part of the fuel accompanied by a high level of NO_x generation. It was believed that the side arm arrangement would overcome the difficulties of a 100-percent premixed burner. In order to convert Configuration 6 to Configuration 13 (an ultra-lean burner), only the ratio between the head end air flow (B-4 + side arm) and the rest of the burner air flow (quench + dilution) had to be shifted.
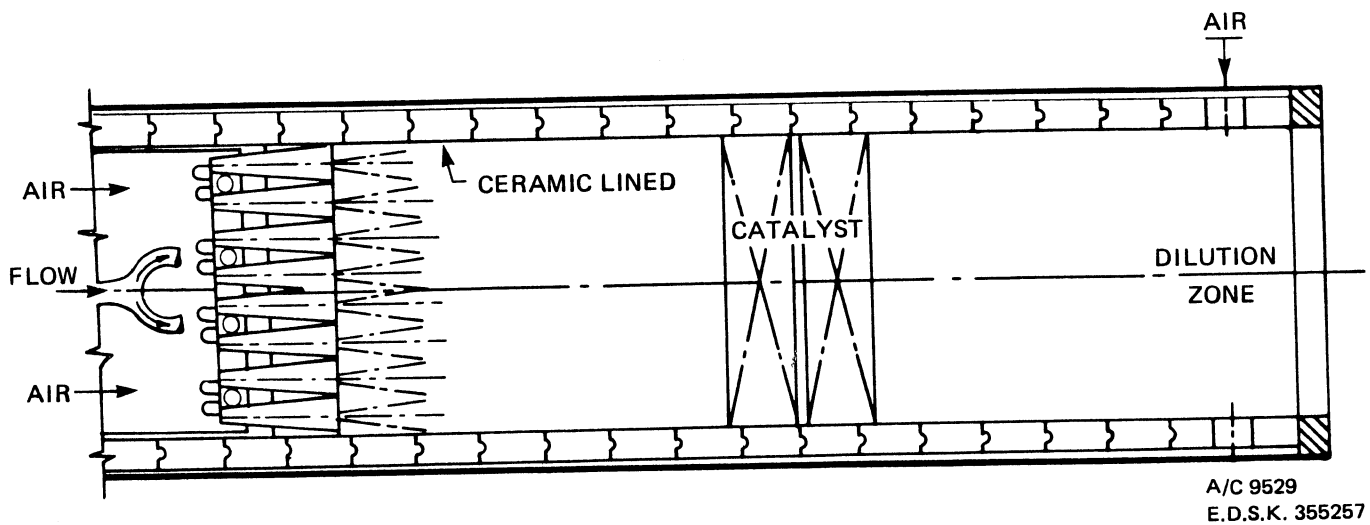
Figure 3-16. Combustor Assembly (Conical Tube Fuel Injector)
Catalytic Configuration

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Section 4
TEST FACILITY

This section describes the four sections of test facility:

- Test Rig
- Fuel System
- Instrumentation
- Data Acquisition System

4.1 TEST RIG DESCRIPTION

The test rig used in this program is shown in Figure 4-1. The rig pressure boundary is formed by flanged sections of stainless steel pipe designed for 2.07 MPa (300 psig) at a 538°C (1000°F) surface temperature. The rig is comprised of three major parts: inlet section, combustor section, and pressure control/exhaust section.

The inlet section consists of the valve and piping arrangement that directs unvitiated primary combustion air (preheated to approximately 371°C/700°F) to the combustor section. The vaporizer portion of the inlet section is a smooth-walled duct. Flow perturbations are minimized in this section to avoid flashback and autoignition. When prevaporized/premixed configurations are tested, fuel is sprayed into the primary air stream to allow vaporization before combustion.

The combustor section is made up of three subsections: rich burner, quench, and lean burner. Exit gas from this section simulates nominal turbine inlet conditions of 1149°C (2100°F) and 1.24 MPa (180 psig).
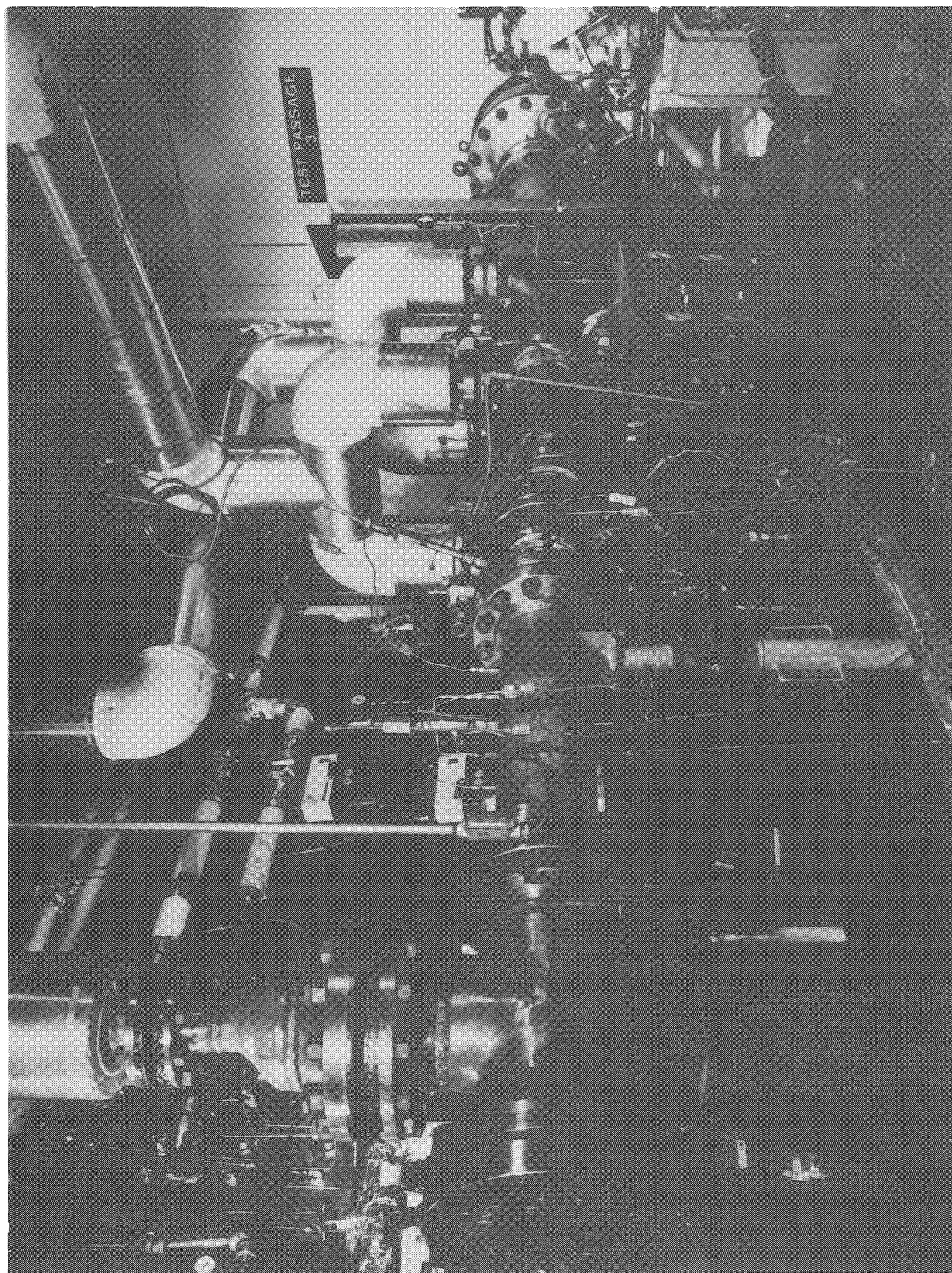
Figure 4-1. High Pressure Combustor Test Passage

The basic design approach is to standardize flame tube geometry and change internal combustor details to yield different configurations. This may be done by independently removing and replacing combustor subsections. While a particular configuration is tested, another can be modified with the "plug in" components for subsequent testing.

4.2 FUEL SYSTEM

The petroleum distillate fuel is stored in a 10,000-gallon underground tank and transferred to 500-gallon day tanks in the fuel blending building. These tanks are connected to the combustion test passage.

Petroleum residual fuel oil is stored in a 5,000-gallon heated storage tank and is piped to a 1,000-gallon heated and insulated day tank in the fuel blending building. It is then pumped to the test passage through heated and insulated piping.

SRC-II coal-derived liquid fuel is stored in an underground 5,000-gallon tank and transferred to a 500-gallon day tank in the fuel blending building. The day tank is connected to the combustion test passage.

Pyridine is stored in a 55-gallon drum and pumped by a separate metering pump to the combustion test passage.

The fuels are blended and mixed on-line, and their flow ratio is controlled automatically. Figure 4-2 shows the piping arrangement of the fuel blending system.
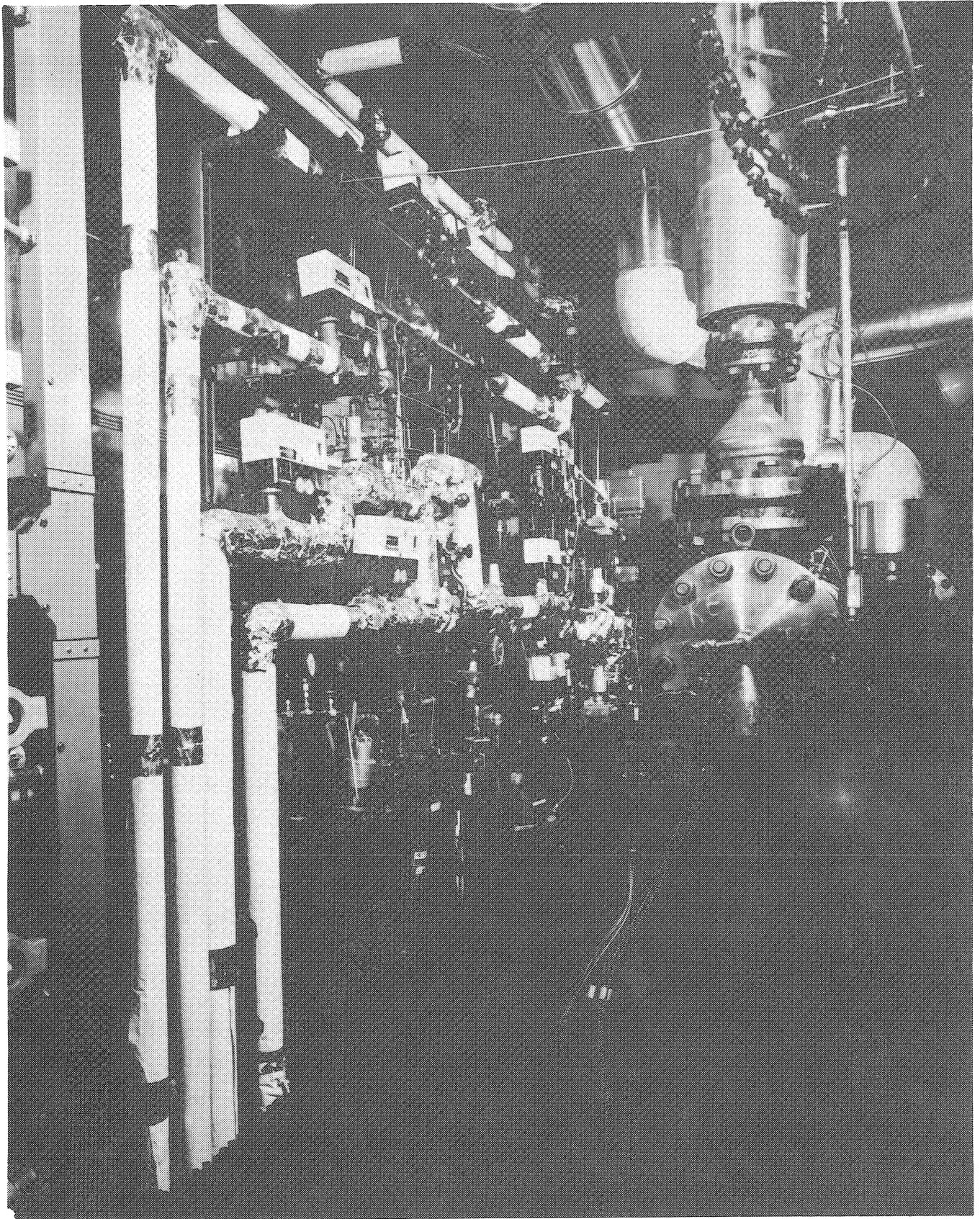
Figure 4-2.   Fuel Blending System

4.3 INSTRUMENTATION

Instrumentation is provided to monitor rig inputs and determine rig section and overall performance. Flow rate, temperature and pressure are measured for fuels, pyridine, primary air, secondary air, atomizing air and cooling air. Atomizing air and fuel pressure drops are measured across the fuel nozzles. Inlet air humidity is also determined in the compressor exit.

Test rig performance is determined by instrumentaion stations as shown in Figure 4-3. Each station consists of a water-cooled 127 mm (5-inch) unit housed in a 203 mm (8-inch) flange. There are multiple penetrations for a static pressure tap, thermocouples that measure temperature, and water-cooled sampling probes that analyze emissions. There are flanged instrumentation stations at the rich burner outlet, the quench module outlet, and the lean burner outlet. This arrangement allows each combustor component to be individually evaluated. Rig duct metal surface and ceramic internal temperature measurements reduce the possibility of burnout and indicate both radial and axial temperature profiles. Surface-mounted thermocouples measure metal surface temperature.

All of these parameters are continuously monitored. Selected variables are recorded on strip chart recorders and trend recorders. All data is fed to the data acquisition system.

Figure 4-4 shows the relative location of the instrumentation in the various sections of the test rig.

On-line continuous monitoring of combustion rig emission is accomplished with a Beckman gas analysis system. The system includes a heated sample line, a smoke meter with detached densitometer, and the instruments in the following list.
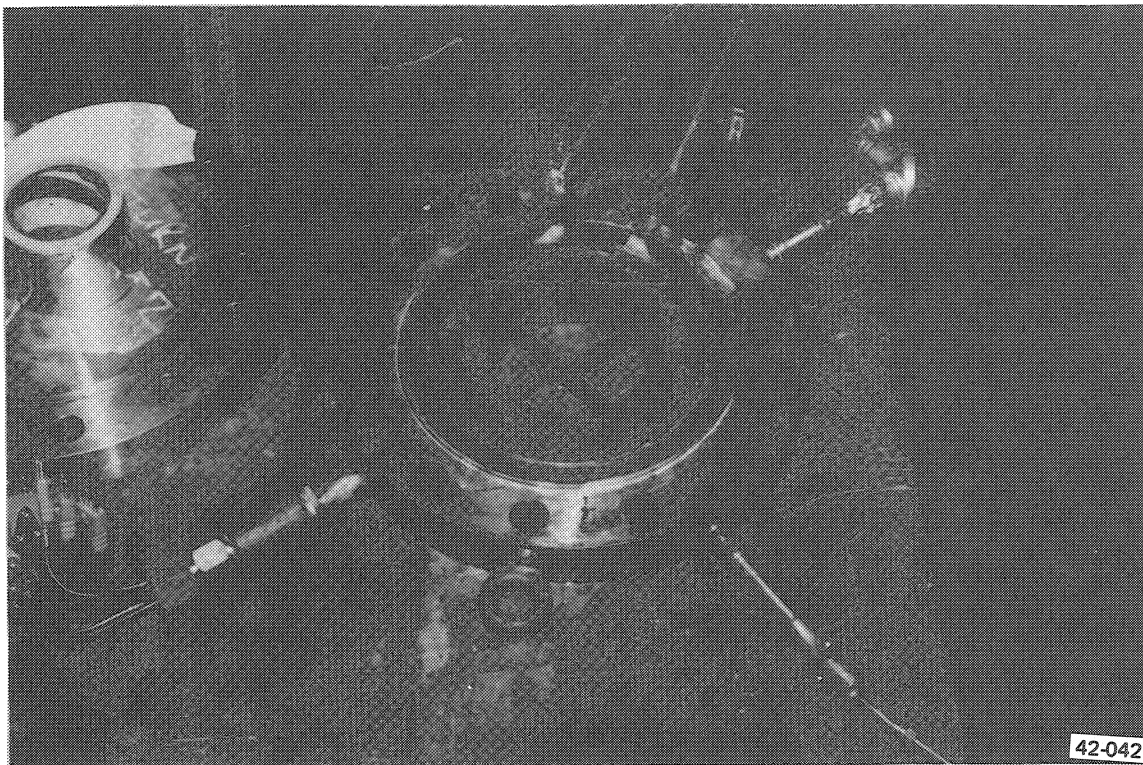
Figure 4-3.   Instrumentation Flange Showing Temperature Rake and Emission Sampling Probe
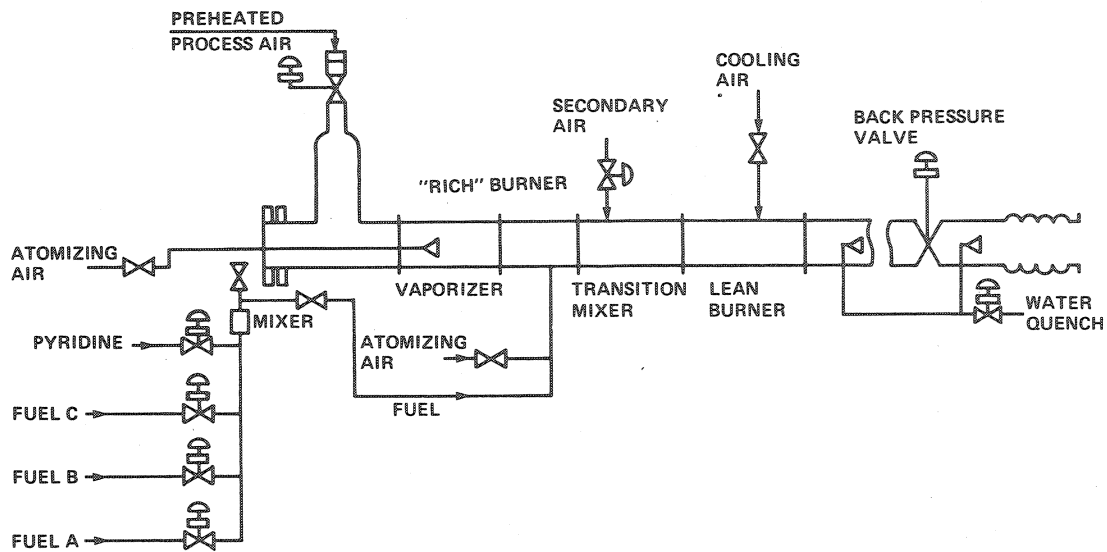
Figure 4-4.   Process Flow Diagram

ANALYZER	METHOD	TYPE
NO/NOx	Chemiluminescence	Beckman 951 H
CO	Infrared	Beckman 865
CO_2	Infrared	Beckman 865
H_2O	Infrared	Beckman 865
O_2	Magnetic susceptibility	Beckman
Hydrocarbons	Flame Ionization	Beckman 402
NOx	Electrochemical	Dynasciences
Smoke	SAE	

All parameters, except smoke, are recorded locally, transmitted to the data acquisition system and put on magnetic tape. Sample flow to any instrument can be interrupted for on-line calibration checks, and all calibrations are checked before and after each test period.

4.4 DATA ACQUISITION SYSTEM

A PDP 11/34 computer system supports data acquisition. This system includes 128K words of MOS memory, 7.5 million bytes of mass storage (disk) capacity, magnetic tape drive, video CRT, hard copy terminal, a line printer and an interface to process instrumentation through A/D converters with multiplexers and digital sensor inputs.

Raw data is collected at regular intervals, analyzed for high/low limits to generate appropriate alarms, and reduced on line to generate various flows, averages and fuel/air ratios. Raw data and reduced data, including all process constants needed for data reduction, are stored on magnetic tape for permanent off-line data retrieval and further data reduction.

The CRT display and printer provide continuous monitoring of the system during a test.

Section 5
FUEL CHARACTERISTICS

Three fuels are used in the test program: petroleum distillate, petro-
leum residual, and a coal-derived liquid. The petroleum distillate
selected as the NASA research test fuel is known as an Experimental
Referee Broadened Specification (ERBS) aviation turbine fuel (1). It
is a blend of 65 percent kerosene and 35 percent hydrotreated catalytic
gas oil. The exact blend of the composition is determined by the
hydrogen content required; the bound nitrogen content is low.

The fuel analyses for all three fuels are given in Tables 5-1 and 5-2.

Table 5-1

FUEL PROPERTIES

	ERBS	Petroleum Residual	SRC-II Middle Distillate
Gravity, °API (15.6°C 60°F)	38.2	15.5	13.3
Specific Gravity	.8388	.9626	.9772
Hydrogen, wt %	12.55	11.43	9.19
Nitrogen, wt %	.008	.22	.8
Sulfur, wt %	.09	.48	.25
Carbon, wt %	87.39	87.03	86.31
Oxygen, wt %	< .5	1.01	1.62
Proton Type, % of Total			
Aromatic	10	9.5	40
Olefinic	10.5	11	31.5
Saturates	79.5	79.5	28.5
Ash wt %	< .0002	.03	.0015
Pour Point °C (°F)	-45.6 (-50)	23.9 (+75)	-59.4 (-75)
Water, wt %	None	trace	trace
Sediment, wt %	trace	50	trace
Viscosity, cst @37.8°C (100°F)	1.87	2653	4.03
Halogens			
Cl, ppmw	<4	35	67
Floride, PPmw	< .2	.65	< .2
Distillation Temp. °C (°F)			
IBP	124 (255)	180 (356)	112 (234)
10%	180 (356)	250 (482)	195 (383)
50%	224 (435)	358 (676)	242 (486)
FBP	400 (752)	480 (896)	360 (680
Net Heat of Combustion			
Btu/Lb	18, 343	17,609	16,674

Table 5-2

TRACE ELEMENTS IN PARTS PER MILLION

	ERBS	Petroleum Residual	SRC-II MIDDLE Distillate
Aluminum	0.4	1.2	1.2
Silver	< 0.1	< 0.1	< 0.1
Boron	< 0.1	0.1	0.1
Barium	< 1.	12	< 1.
Beryllium	< 0.1	< 0.1	< 0.1
Bismuth	< 0.1	< 0.1	< 0.1
Calcium	< 0.4	20	< 0.4
Cadmium	< 0.1	< 0.1	< 0.1
Cobalt	< 0.1	1.	< 0.1
Chromium	< 0.1	0.25	1.
Copper	< 0.1	0.4	12
Iron	1.2	12	12
Germanium	< 0.1	< 0.1	< 0.1
Magnesium	0.4	4.	0.1
Manganese	< 0.1	0.4	0.1
Molybdenum	< 0.1	< 0.1	< 0.1
Nickel	< 0.1	8	0.1
Potassium	< 1.	20	< 1.0
Lead	2.5	0.5	0.6
Antimony	< 0.4	< 0.4	< 0.4
Silicon	1.	12	1.2
Sodium	< 1.	60	< 1.
Tin	< 0.1	< 0.1	1.2
Titanium	< 0.1	0.6	0.1
Vanadium	< 0.1	8.	< 0.25
Zinc	< 0.6	4.	0.6

REFERENCE

1. Prok, G.M., and Seng, G.T., "Initial Characterization of An Experimental Referee Broadened - Specification (ERBS) Aviation Turbine Fuel," NASA Tech. Memo 81440, January 1980.

Section 6

COMBUSTOR TEST RESULTS

This section presents the test results for Configurations 1, 2, 4, 6, 7, 8, 9, 10, 11, 12 and 13. The results are presented in significant detail where appropriate.

6.1 CONFIGURATION 1 - DIRECT INJECTION, VENTURI QUENCH

To find the ignition characteristics of the modified B-4 type burner, it was first tested in an atmospheric pressure rig with ERBS fuel. At an estimated air flow of 0.14 Kg/sec at 65°C, ignition occurred at a lean equivalence ratio (ϕ) of 0.270 with a bright yellow flame outside the burner. At this air flow, fuel flow was decreased to obtain lean blowout and increased to obtain rich blowout. Following is a summary of the results.

OIL FLOW M^3/min($\times10^4$)	AIR FLOW Kg/SEC	FUEL/AIR RATIO f/a	ϕ_p	COMMENTS
0.05	0.14	0.0187	0.27	Ignition
0.03	0.14	0.0112	0.16	Lean Blowout - Blue Flame
0.10	0.14	0.0374	0.54	Yellow Flame
0.14	0.14	0.0524	0.76	
0.20	0.14	0.0749	1.08	
0.32	0.14	0.120	1.73	Yellow Fluffy Flame
0.42	0.14	0.157	2.27	Some Soot Observed

These descriptions are for the flame outside the rich burner, which includes the combustion products in the rich burner that react with ambient air.

The data shows that the Configuration 1 rich burner appears to be stable over a wide range of fuel/air ratios. The data also shows that the program target $\phi = 1.7$ was easily obtained.

Configuration 1 was tested at high pressure with ERBS fuel, with ERBS fuel and pyridine to simulate the fuel-bound nitrogen, and with SRC II fuel oil (coal-derived liquid). There were no problems with ignition or during switch-over to rich/lean operation.

Results for the full pressure conditions are as follows:

ERBS FUEL TESTS

The full pressure, full fuel runs shown in Figure 6-1 for ERBS fuel were conducted under the following conditions.

<div align="center">

Nominal Test Conditions, Configuration #1

</div>

Primary Air Flow, Kg/sec	0.32
Total Air Flow (Primary + Quench + Dilution), Kg/sec	1.8
Fuel flow, $M^3/min(x10^4)$	30.0
Burner Outlet Pressure, MPa	1.3
Burner Outlet Temperature, °C	1065.0
Burner Inlet Temperature, °C	315.0

The burner outlet rather than inlet pressure was included because the control system was coordinated with the burner outlet pressure. This arrangement was considered advantageous because the chemical processes in the combustor take place primarily at the exit pressure level. In some of the experimental configurations, this level differed significantly from the inlet pressure level.

The measured NO_X values are corrected to 15 percent O_2 and are plotted in Figure 6-1 as a function of the primary zone equivalence ratio, ϕp. The
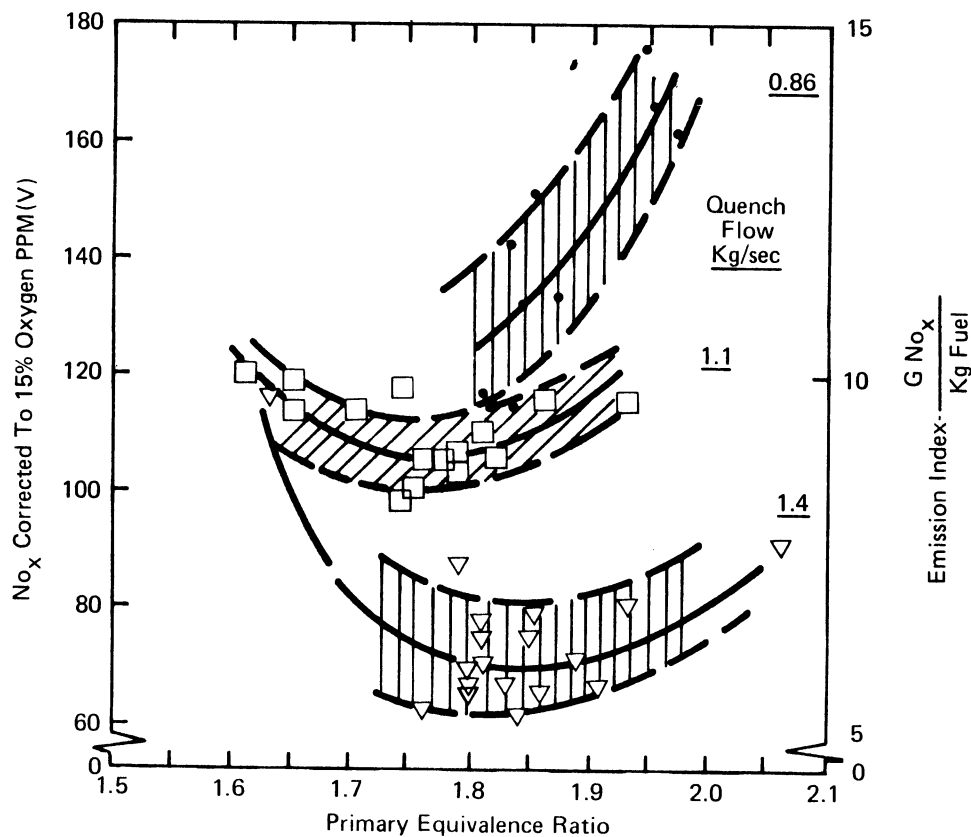
Figure 6-1.   NOx (corrected to 15% O_2) for Rich-Lean
              Configuration 1 and ERBS Fuel

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secondary variable was the quench flow, M_Q (Kg/sec). Three values of M_Q
were used in this test series.

The test results show that the NO_x emissions of the burner have a minimum
value of 1.7 to 1.85 at a primary equivalence ratio, ϕp. The data shows
approximately ±10 percent scatter about the average. This distribution
is caused by:

●   Varying geometry of the test passage due to erosion of the
     ceramic test sections

●   Expected standard variations in air and fuel temperature,
     pressure, nozzle performance, instrumentation and other related
     conditions

Varying the amount of quench air had a significant effect on the level of
NO_x generated; larger quench flows resulted in lower NO_x values. It is
not clear from the data whether the absolute amount of quench flow or the
ratio of the axial flow from the rich burner to the quench flow governs

6-3

NO_x generation. In one test in the series, the quench flow was maintained at 1.1 kg/sec and the primary air flow was reduced from 0.32 kg/sec to 0.23 kg/sec; these conditions caused a decrease in NO_x emissions from ~105 ppm to ~70 ppm. This result suggests that the flow ratio is the influencing variable.

Post-test inspection of the hardware, primarily the downstream end of the rich burner and the throat of the quench section, suggested that the opposing jets of the quench flow created a recirculation region upstream of the quench section. The recirculation region created an unwanted reaction zone between the unburnt fuel and the quench air. This condition caused a high temperature combustion zone, which in turn damaged the ceramic walls and changed the geometry of the test section.

It was thought that the original design diameter of the venturi quench throat was conducive to creating such a recirculation region. If the throat region was smaller, the damage may have been minimized or avoided. During the course of the tests, the NO_x levels dropped; this indicated that the melted ceramic had narrowed the quench throat diameter, which may have eliminated or reduced the unwanted combustion region. Toward the end of the program an attempt was made to test the performance of a smaller throat quench section; however, inadequate curing of the ceramic section made it impossible to obtain a definitive answer to the question of long term geometrical stability.

CO VS. NO_x AND NO_x FORMATION

The NO_x results (Figure 6-1) suggest that the quench flow determines the absolute level of minimum NO_x. When the quench flow was increased from 0.86 kg/sec to 1.4 kg/sec, the minimum NO_x was reduced from ~110 ppm to ~70 ppm. To determine if additional increases in quench flow would result in an additional decrease in NO_x, it is necessary to know if the temperature in the quench section is high enough to generate thermal NO_x.

Since it is difficult to measure a representative average gas temperature in the quench section, the temperature was calculated for each test
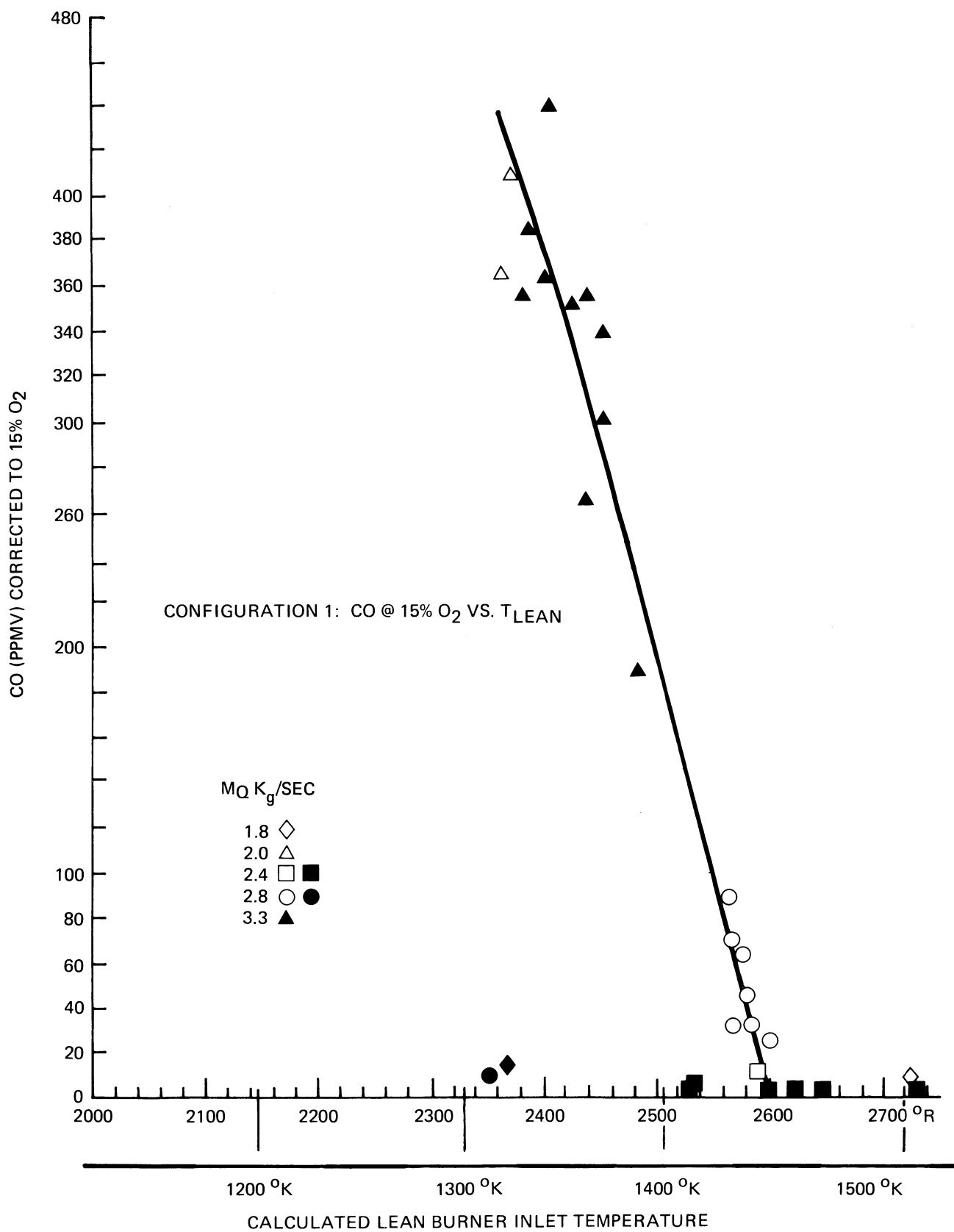
point. The temperature was calculated at the downstream end of the quench section (T_{Qlean}) by subtracting the cooling effect of the dilution air, and at the upstream end (T_{Qrich}) by reducing the calculated rich burner exit temperature with the injection of dilution air. Although each method gave a different result, T_{Qrich} was always higher. The differences were not significant in terms of estimating the temperature in the quench section; and they were due to uncertainty in measuring the dilution air.

Figure 6-2 shows that the CO results provide a way to estimate T_{Qlean} for Configuration 1. It appears that above 1170°C (2600°R), no CO is generated. Figure 6-3 is a plot of CO vs. NO_x results for various quench flows. It is reasonable to assume that the quench temperature, as represented by the CO levels, is the significant NO_x-creating variable. When CO is lower than approximately 40 ppm (Figure 6-3), NO_x increases rapidly. When CO is above 40 ppm, there is a constant level of NO_x (~70 to 80 ppm). From Figures 6-2 and 6-3 it can then be concluded that if the quench temperature is kept below ~1170°C, CO remains high and no thermal NO_x is generated in the quench section. The following conclusions are drawn from the NO_x data for Configuration 1:

- The NOx generated in the rich section is a function of φp and is a minimum in the range 1.7-1.85.

- Increasing the quench flow decreases the minimum value of NOx generated to a value of ~60-70 ppm. Further increases in quench flow which will decrease for quench temperature to <1170°C will have little effect on NOx.

- The 60-70 ppm NOx coming from the rich burner may be reduced by improvements in burner design.

COAL-DERIVED LIQUID (CDL) TESTS

NO_x results corrected to 15 percent O_2 for the CDL tests of Configuration 1 are presented in Table 6-1. Comparable results are presented for ERBS fuel tests in order to calculate the conversion of fuel-bound nitrogen (FBN) to NO_x.

Figure 6-2. Configuration 1: CO at 15% O_2 vs. T_{QLEAN}

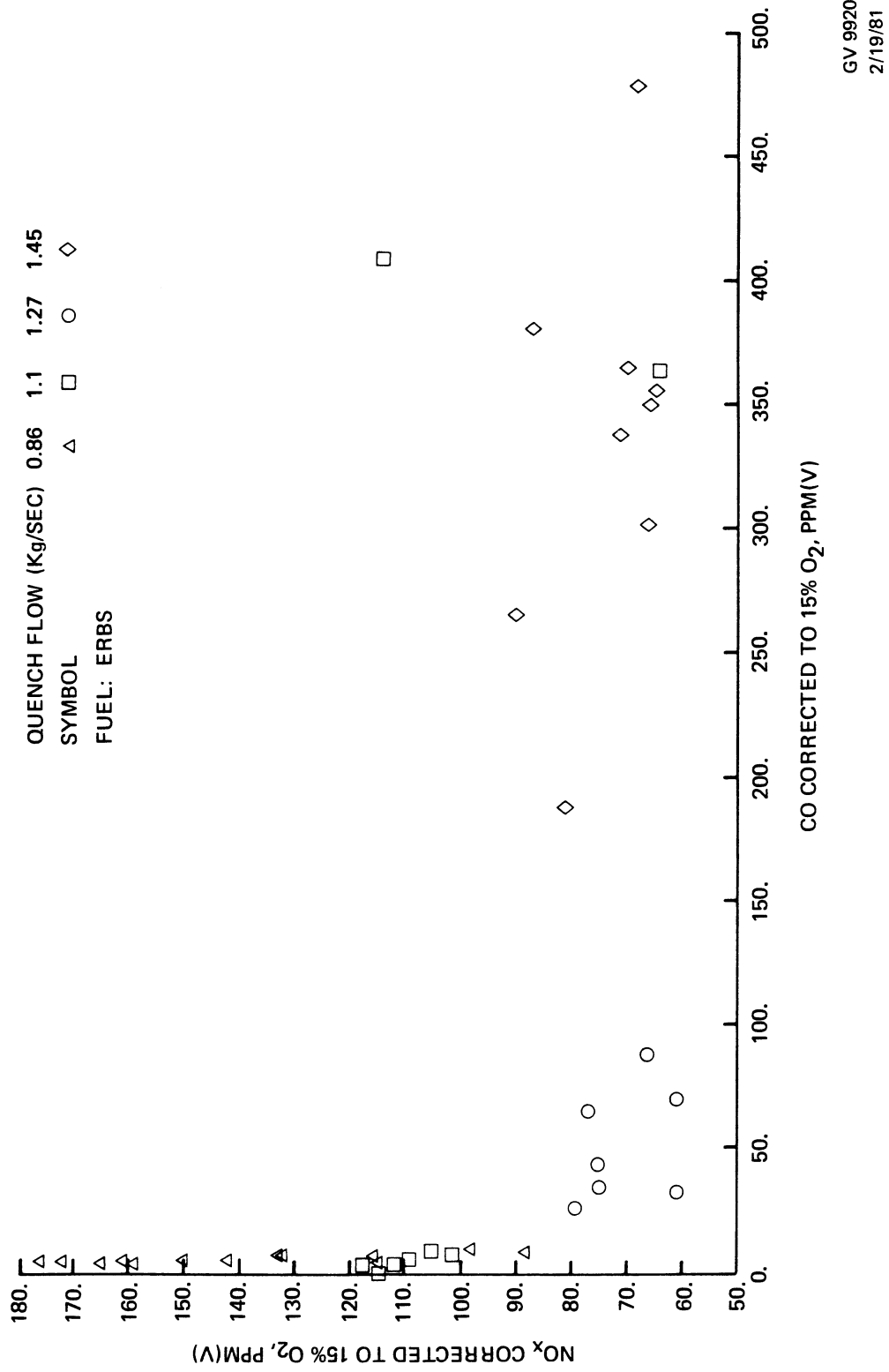QUENCH FLOW (Kg/SEC) 0.86 1.1 1.27 1.45
SYMBOL △ □ ○ ◇
FUEL: ERBS

CO CORRECTED TO 15% O₂, PPM(V)

NO_x CORRECTED TO 15% O_2, PPM(V)

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Figure 6-3. CO vs. NO_x - Configuration 1

6-7

Table 6-1

NOx RESULTS, CDL VS. ERBS FUEL

Fuel	M_Q (kg/s)	ϕ_p	NOx at 15% O_2 ppm	Δ NOx ppm	100% FBN NOx ppm	% Conversion
CDL	1.1	1.7	169	63	339	18
ERBS	1.1	1.7	106			
CDL	1.2	1.7	170	50	351	14
ERBS	1.2	1.7	120			
CDL	1.2	1.8	147	47	340	14
ERBS	1.2	1.8	100			

ΔNO_x is the difference between the CDL and ERBS fuel results for the same
test conditions. (ERBS results are taken from Figure 6-1.) Assuming
that all (0.8 w/o) the FBN is converted to NO_x, 100 percent FBN NO_x is
the value calculated. The conversion efficiencies (ΔNO_x/100% FBN NO_x) of
14 to 18 percent compared well with those measured by Sarofim (1) at
similar levels of FBN. It is interesting to note that his results were
obtained at primary equivalence ratios on the order of 1.2 to 1.3.

ERBS FUEL WITH PYRIDINE TESTS

A series of tests were run with ERBS fuel doped with pyridine (C_5H_5N) in
order to determine the conversion efficiency of a simulated FBN fuel.
Some difficulties were encountered with the control system, and the
pyridine injection rates are questionable. In the test run closest to
the CDL conditions, a conversion efficiency of 10 percent was measured,
which is lower than for CDL FBN (14 to 18 percent). Whether this dif-
ference is due to a different chemical conversion process for pyridine or
inaccuracies of measurement cannot be determined.

RESIDUAL FUEL TESTS

A series of tests were run with a residual petroleum fuel containing 0.22
percent FBN. When compared to similar ERBS results, a conversion
efficiency of 30 percent was obtained (FBN into NO_x). When compared to

CDL results, the conversion efficiency is higher. This is expected because the residual oil contains less FBN than CDL, so there are fewer nitrogen compounds competing for the available O_2.

COMBUSTION EFFICIENCY, SMOKE AND UNBURNT HYDROCARBONS

All tests discussed in conjunction with the NO_x results had a combustion efficiency greater than 99.5 percent (calculation of the efficiency was based on the exhaust gas analysis). The efficiency was not materially influenced by the type of fuel used, as shown in Table 6-2 for some characteristic test points.

Table 6-2
COMBUSTION EFFICIENCY, CONFIGURATION 1

Fuel Type	ERBS	ERBS	ERBS	CDL	CDL
Overall Fuel/Air Ratio	0.025	0.022	0.019	0.022	0.019
Combustion Efficiency %	99.97	99.97	99.96	99.76	99.51
ϕ_p	1.83	1.84	1.95	1.73	1.58

Two smoke measurements were taken from Configuration 1 during ERBS fuel operation, and two more during SRC II fuel oil operation. The results are shown in Table 6-3.

Table 6-3

CONFIGURATION 1 SMOKE RESULTS

Fuel	ϕR	M_Q Kg/sec	SAE Smoke #
ERBS	1.81	1.5	26
ERBS	1.71	1.1	20
CDL	1.58	1.1	29
CDL	1.81	1.2	27.5

In addition to the smoke measurements in the rich/lean mode reported above, smoke measurements were taken when Configuration 1 was operated as a lean burner (preparation for changeover to rich/lean operation). There was little discoloration of the smoke patches under lean conditions.

The SAE smoke numbers indicate that there will have to be a way to reduce the smoke produced by Configuration 1 if it becomes the object of further development. No optimization of smoke was attempted since NO_x was the primary interest in this study. The 20 to 29 SAE number range should be considered in conjunction with the SAE 20 goal of the program. There were not enough smoke tests performed on this configuration to determine the smoke generation mechanism in this burner.

In all the test runs of Configuration 1, unburnt hydrocarbons in the exhaust were found to be 7 to 20 ppm, corrected to 15 percent O_2. This level is not significant.

The measured average burner outlet temperatures were calculated from six thermocouple readings downstream of the dilution section. In the test runs discussed, the averages ranged from 1100°C to 940°C on ERBS fuel and from 990°C to 935°C on CDL. The profile factor for the two ERBS runs was 0.7 percent and 1.3 percent; for the CDL runs it was 3.0 percent and 2.9 percent.

Fuel/air ratios based on the oxygen content of the exhaust as well as CO_2 were calculated. In the tests discussed here, the obtained values ranged from 0.024 to 0.015. Of the CO_2-based and O_2-based fuel/air ratios, the O_2 ratio was usually higher by a difference of one in the third digit (0.021 vs. 0.020).

The overall fuel ratio was also calculated from the measured fuel and air flow $(F/A)_0$. Based on $(F/A)_0$ and $(F/A)_{CO2}$, the fuel/air ratio-ratio (FARR) was obtained. For the test runs discussed here, FARR was generally between 0.92 and 1.0; however, in three runs FARR was approximately 0.89, and in four runs FARR was between 0.77 and 0.88.

PRESSURE DROP

The geometry of the Configuration 1 quench section changed during testing; therefore, pressure drop is discussed in terms of the nominal diameter quench section. The data shown in Figure 6-4 was first plotted for quench flow M_Q = 1.1 Kg/sec, which was the quench flow with the most runs available. Since $\Delta P/P$ is approximately proportional to ΔT, the linear relationship was used.

Since the pressure drop in Configuration 1 is dominated by the quench flow, it was assumed that by changing M_Q, $\Delta P/P$ will change approximately as a square relationship (assuming that the absolute pressure is about constant). The other two M_Q = const. lines were drawn accordingly.

In some runs, the pressure drop unexpectedly departs from the expected level. It is probable that changes in the throat geometry were responsible for the increase in $\Delta P/P$.

SUMMARY OF CONFIGURATION 1

The rich/lean configuration is capable of meeting EPA requirements for both the ERBS and residual fuels. It also comes very close to meeting the requirement for CDL (150 to 170 ppm vs. 140 ppm objective). In its present design the rich burner produces approximately 60 to 70 ppm NO_x at

ϕ_p in the range of 1.7 to 1.85. Further NO_x reduction may be possible with a better pre-mix arrangement. Configuration 1 is a burner that should be optimized in any future programs.
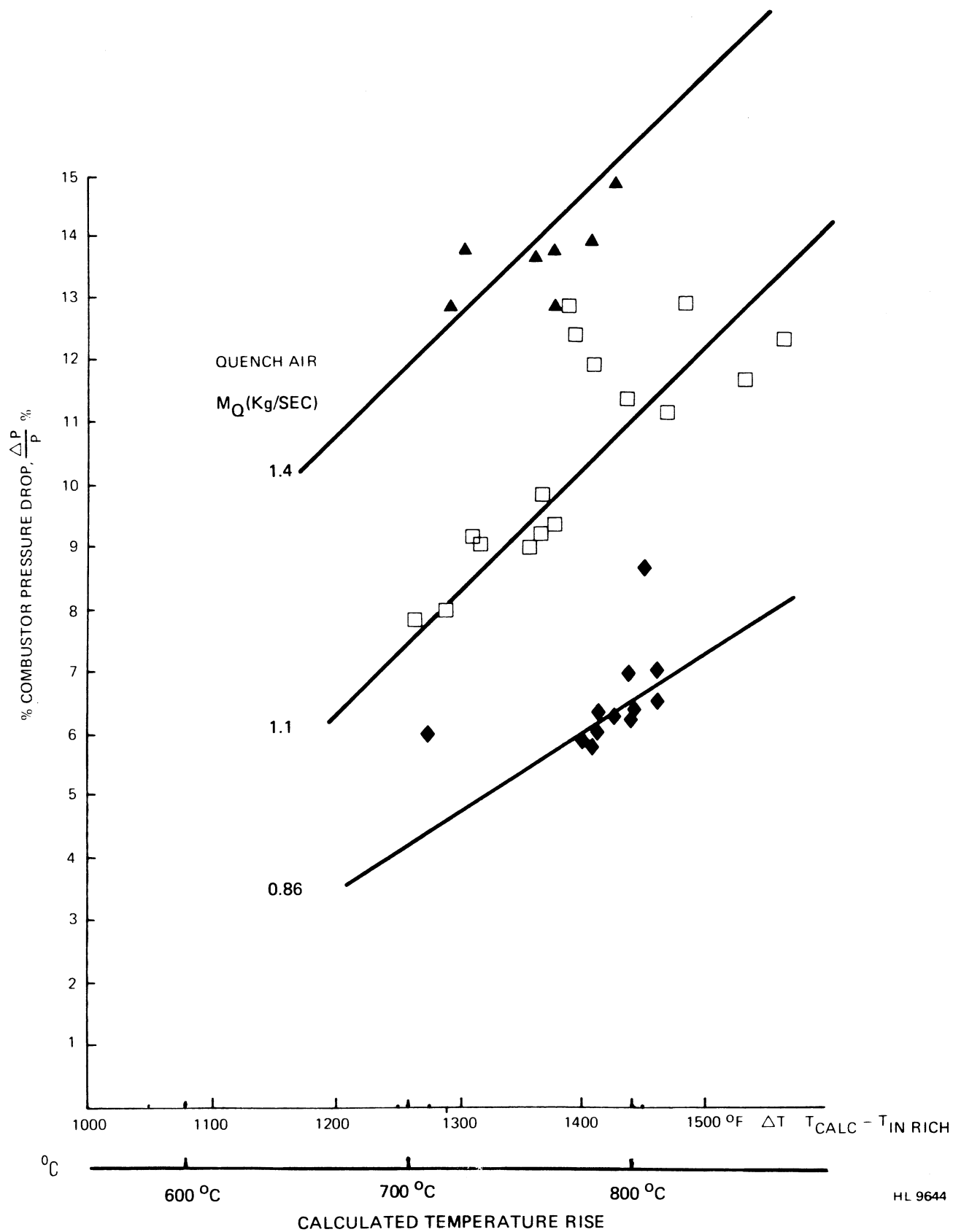
Figure 6-4.   Pressure Drop vs. ΔT - Configuration 1

6.2 CONFIGURATION 2 - DIRECT INJECTION, VORTEX QUENCH

As described in Section 3.2, Configuration 2 consists of the same rich
burner, straight pipe lean burner, and dilution modules as Configuration
1. The venturi-type quench of Configuration 1 was replaced by a two-disk
vortex mixer. There were two test series for this configuration. During
the first series, the upstream disk of the vortex mixer was operated with
its tangential slots open; the downstream disk was used with its
secantial slots in operation. This was intended to make the quench flow
penetrate to the centerline of the second swirler; it was also an attempt
to stop the swirl of the first swirler by providing counter rotation.
During the second series, the upstream disk was operated secantially for
reasons explained below (Configuration 2A).

NO_x RESULTS

The nominal conditions are shown in Table 6-4.

<div align="center">

Table 6-4

NOMINAL TEST CONDITIONS, CONFIGURATION 2

</div>

Primary air flow, Kg/sec	0.27
Total air flow, Kg/sec	1.7
Fuel flow, $M^3/min(x10^4)$	26.0
Burner outlet pressure, MPa	1.1
Burner outlet temperature, °C	1070.0
Burner inlet temperature, °C	325.0

The NO_x results, corrected to 15 percent O_2, are shown in Figure 6-5.
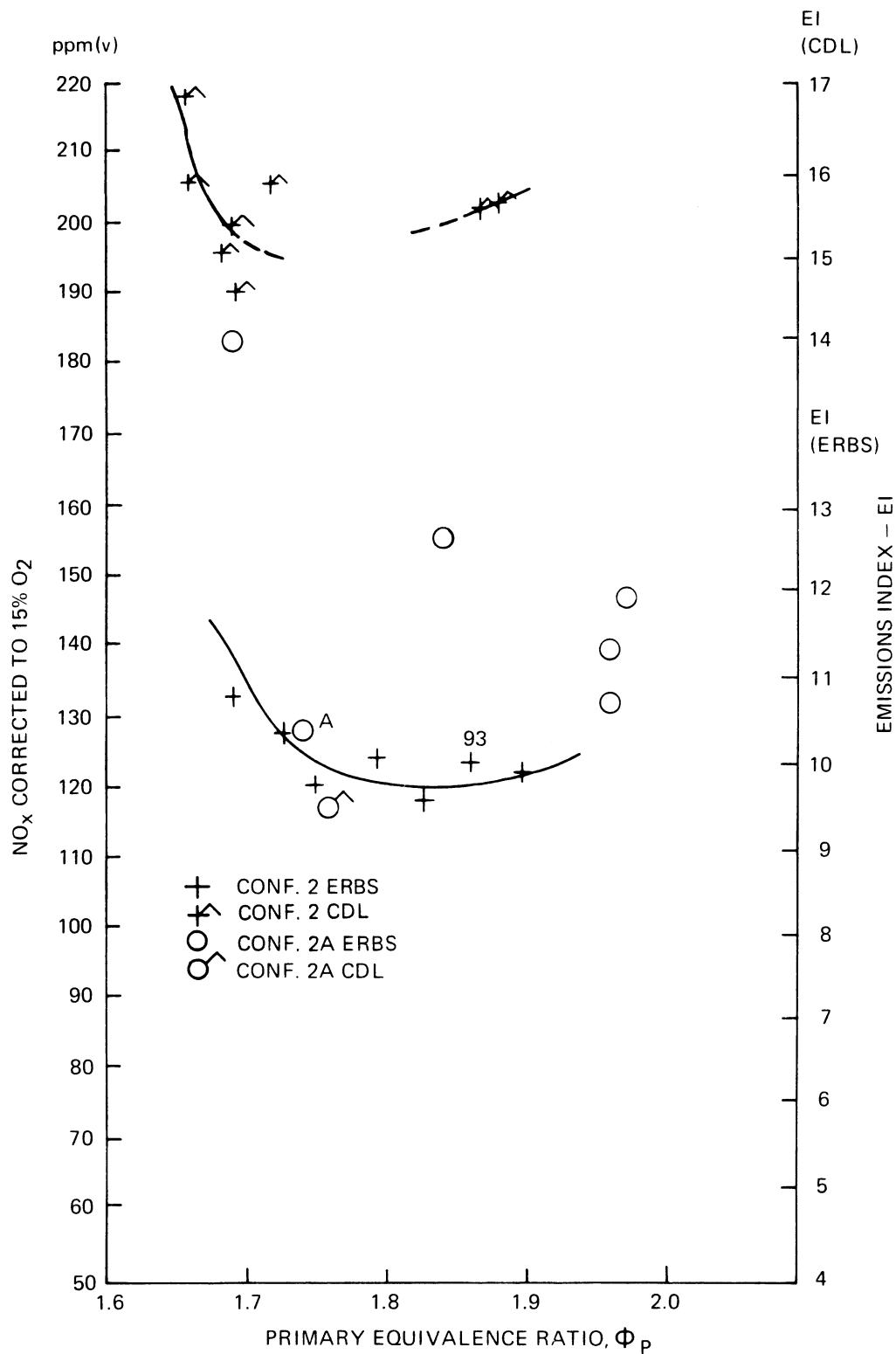The independent variable is the rich (primary) burner equivalence ratio,
ϕ_p.

Figure 6-5.   Configuration 2: NO_x versus Primary Equivalence Ratio

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The quench flow, (M_Q), was not varied in testing Configuration 2. For the ERBS fuel tests, M_Q was 1.13 Kg/sec and for CDL fuel M_Q was 1.23 Kg/sec.

The ERBS NO_x results show the same trend as in Configuration 1 with a minimum occurring at ϕ_p ~1.8. However, the minimum NO_x level is higher (120 vs. 100 ppm) than Configuration 1 for the same quench flow. The additional NO_x generated with CDL containing FBN is about the same as Configuration 1, 60 to 70 ppm. This result was expected as the rich portion of the burner was identical in both configurations.

OTHER EMISSION AND PERFORMANCE RESULTS

In the process of testing Configuration 2, an instrument malfunction occurred and no CO data was obtained. Unburnt hydrocarbons were measured in the 9 to 15 ppm range, which is considered insignificant. Smoke measurements were about the same or higher than Configuration 1, and no attempt was made to reduce the smoke in these screening runs.

The overall fuel/air ratios, calculated from the fuel and air flows, ranged from 0.018 to 0.023 for ERBS and CDL fuels. The oxygen-based calculation of the overall fuel/air ratio agreed with the flow calculated value. Because of a lack in CO results, there were no exhaust gas analyses based on combustion efficiencies. The average exhaust gas temperature ranged from 900°C to 1050°C with profile factors ranging from 1.1 to 2.3 percent.

Pressure drops measured during the tests are plotted in Figure 6-6.

MODIFICATION OF CONFIGURATION 2 TO 2A

Although Configuration 2 did not improve the emission levels of Configuration 1, the fact that its metal quench section can be easily modified makes it a candidate for further development.
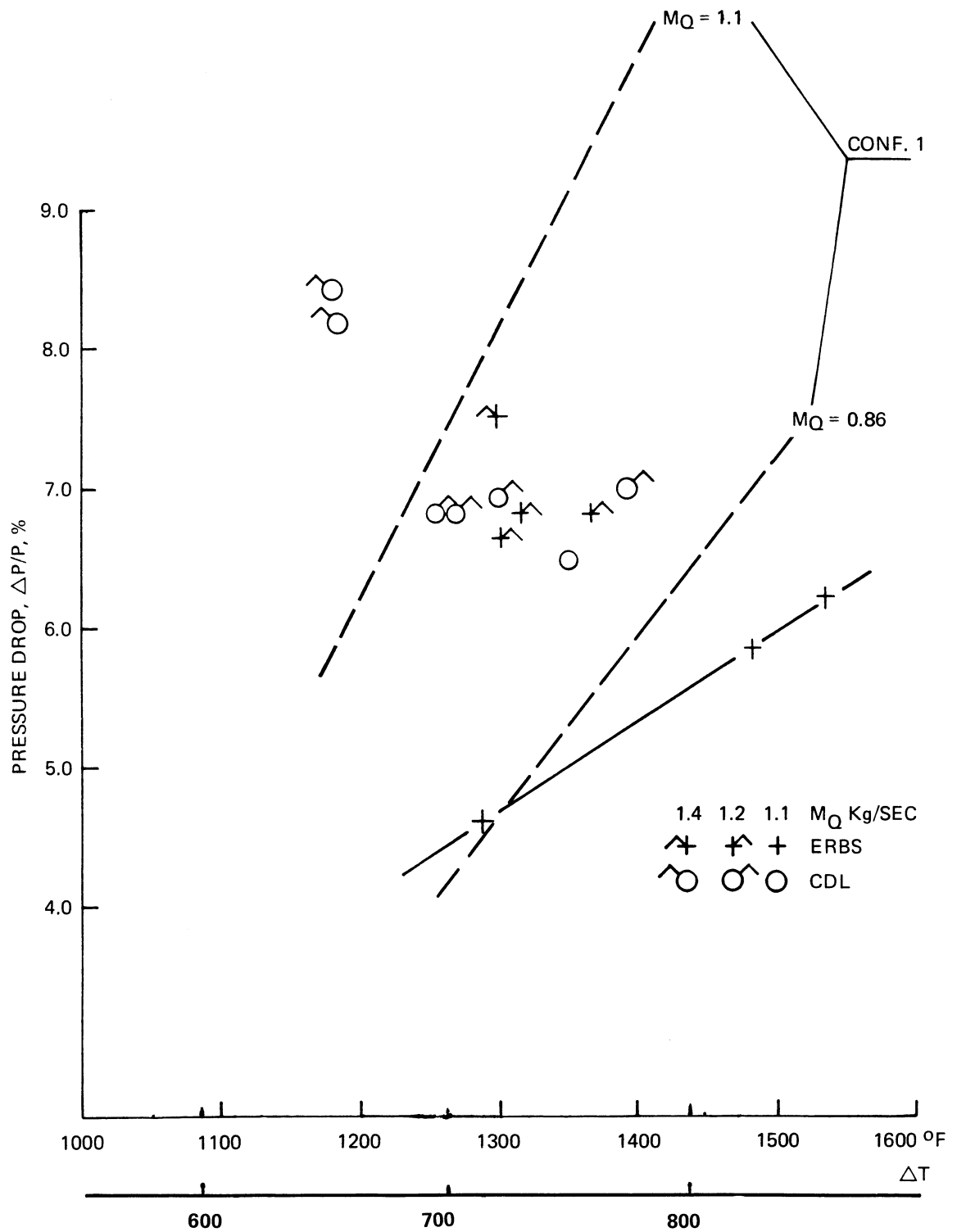
Figure 6-6.   Pressure Drop vs. Change in Temperature

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It was thought that the quench section of Configuration 2 could be modified to bring its minimum NO_x level closer to that of Configuration 1. The modification was a change in the use of the centripetal flow quench section.

In Configuration 2, the first upstream disk of the quench section had eight tangential slots for air admission, and the second disk admitted the air secantially. In the modified version, Configuration 2A, the first disk also was installed secantially. The reason for this arrangement was that the NO_x in Configuration 2 was higher than in Configuration 1, where the quench section admits the quench air through radial holes. Therefore, the secantial quench section was a compromise between radial and tangential air admission. It was expected that the good mechanical performance of Configuration 2 (the metal swirler and the ceramic rich burner both worked quite well) would remain the same in Configuration 2A.

Seventeen test points were obtained at full pressure with Configuration 2A. The NO_x results are shown (Figure 6-5) along with the Configuration 2 data. During the test, the pressure in the burner steadily dropped. Toward the end of the test, it was noticed through the downstream observation port that small pieces of material were carried by the flue gases creating a sparkler effect.

Initially, the NO_x level changed erratically, but eventually it decreased. With ERBS fuel, NO_x increased from 68 ppm to 180 ppm, then decreased to 135 ppm and eventually to 70 to 80 ppm.

Post-test inspection revealed that the ceramic material between the rich burner and the swirler was overheated and gradually blocked the central opening of the swirler. Part of the quench air recirculated into the downstream end of the rich burner, and a local combustion site developed. The resulting high temperature caused a local melting of the ceramic material until the resulting blockage partially eliminated the recirculation.

Configuration 2A geometry began to resemble that of Configuration 1. The reduced cross section, where the hot combustible exhaust of the rich burner meets the fresh air of the quench section, becomes a combustion region and generates NO_x. The resulting ceramic meltdown partially eliminates the recirculation and the NO_x decreases. The NO_x data (Figure 6-5) illustrates this effect.

This phenomenon did not occur in Configuration 2. It is probable that the tangential entry in the first swirler eliminated the recirculation. The NO_x level in Configuration 2, higher than in Configuration 1, indicated that some high temperature combustion could have taken place downstream of the quench swirler. However, judging from the NO_x level (about 135 ppm in Configuration 2 and 183 ppm in Configuration 2A), the combustion site in Configuration 2 may have been less extensive than in Configuration 2A; therefore no structural damage was observed in Configuration 2. These results suggest that the prevention of combustion in the quench module greatly depends on the acceleration of the axial flow of fuel-rich gas before it mixes with the oxidant in the quench section.

CONCLUSIONS FROM THE CONFIGURATION 2 (2A) TESTS

The following conclusions were drawn from test results of Configuration 2:

- Configuration 2 has a higher minimum NO_x level than Configuration 1

- Smoke is comparable to Configuration 1

- Configuration 2 probably has a local combustion site in its quench section similar to Configuration 1. This combustion site may be eliminated by changing the quench flow and/or the arrangement of tangential and secantial slots in the quench section and/or the diameter of the quench flow exhaust

- A successful redesign of the quench section may have a beneficial effect on the durability of the burner and the level of NO_x

6.3 CONFIGURATION 4 - DIRECT INJECTION, VORTEX QUENCH, CATALYST

This configuration includes a rich burn module, a vortex quench module, and a catalytic element in the lean burn module. The catalyst is a design that reduces pressure loss. It is an Engelhard DXE-442 catalyst (Pd/stabilized alumina) on 4 inches of 60 channel per square inch Torvex α-alumina.

The configuration was generally run at an inlet pressure of 1.24 MPa and an inlet temperature of 360°C with an overall fuel/air ratio of 0.019 and a reference velocity of 23 M/S. The data obtained from testing each fuel is explained below.

ERBS FUEL RESULTS

The tests run on ERBS fuel demonstrated NO_x emission of approximately 134 ppmv when corrected to 15 percent oxygen concentration. The equivalence ratio of the rich burner had an apparent effect, and it is possible that inlet air temperature had a slight effect; higher inlet air temperature produced higher NO_x levels. This configuration does not meet the Federal EPA standards for NO_x emission.

SRC-II RESULTS

Tests were also run on Configuration 4 using SRC-II fuel. The NO_x emission was approximately 200 ppmv when corrected to 15 percent oxygen concentration. There was no consistent effect of rich burner equivalence ratio, but it is possible that inlet air temperature had some effect; higher inlet air temperature produced lower NO_x levels.

Figure 6-7 displays the NO_x results of the two fuels at 15 percent oxygen. It was concluded that the conversion of fuel nitrogen to NO_x amounted to approximately 21 percent.
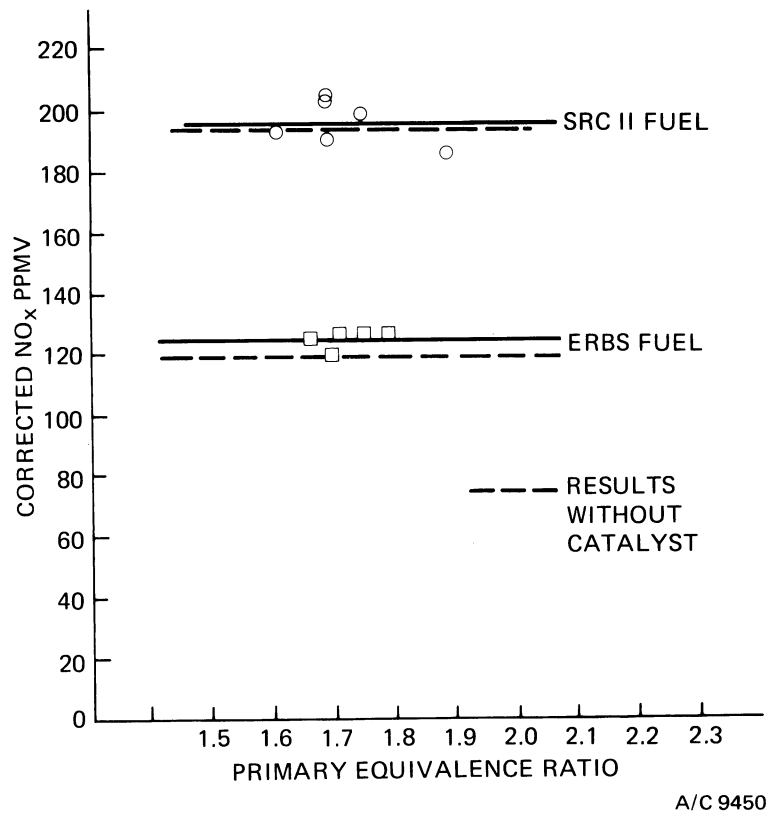
Figure 6-7.   NOx Measurements for Two Fuels with a Catalytic Element in the Lean Burn Module (Config.4) Compared with Configuration 2

CATALYST EFFECT

The test results of Configuration 4 can be compared with those of Configuration 2 in order to evaluate the contribution of the catalyst as a lean burner. The comparison is valid because the only significant difference in hardware between the two combustors is the catalyst. There does not appear to be any difference between the results of the two configurations in terms of either absolute NO_x levels or the conversion of fuel nitrogen to NO_x. The conclusion is that the reaction was effectively complete at the quench module, upstream of the catalyst, so that the catalyst was ineffectual in reducing either the NO_x or the FBN conversions. At these temperatures there is no appreciable reduction in NO_x due to reaction with CO and unburnt hydrocarbons in a manner of the three-way automobile catalysts.

SUMMARY OF CONFIGURATION 4

The most significant findings in testing Configuration 4 were:

- NOx emission for clean fuels did not meet emission goals

- NOx emissons for clean fuel and CDL were essentially invariant with rich burner equivalence ratio

- Conversion of FBN to NOx was approximately constant for all rich burn equivalence ratios at 21 percent

- The catalyst in the lean burner apparently contributed little to reducing NOx or preventing FBN conversion to NOx

6.4 CONFIGURATION 6 - HYBRID PILOTED RICH BURNER, OPPOSED JET VENTURI QUENCH

The modification of the B-4 burner (from conventional to swirler sta-
bilized combustor) made it necessary to test at atmospheric pressure.
Good ignition capabilities were achieved, and the burner (assembled with
the "side arm" and the downstream swirler) was installed in the pres-
surized passage with the other modules of the configuration.

During the first test under pressure, there was no problem with the
piloting B-4 burner: light-off and pressurization were achieved without
unusual difficulties. However, as soon as the fuel flow was started in
the side arm, flashback interrupted the test. The occurrence of
flashback in the side arm was sensed by a thermocouple in the porous
tube. The thermocouple triggered the control system to shut down the air
and fuel flow when the temperature in the porous tube (premixed fuel flow
and air) rose above 550°C.

During the second test, attempts were made to prevent flashback. First,
the system pressure was lowered. With the mass flow the same, the gas
velocity in the swirler (opposed swirl, downstream of the porous tube)
was increased. Another attempt was based on the possibility that
incomplete vaporization in the porous tube causes the raw fuel to ignite
in the swirler. Therefore, the preheated temperature of the combustion
air was increased from approximately 315°C to 388°C. Another attempt to
prevent flashback was to facilitate evaporation with high temperature
atomizing air preheated to 215°C. Eventually, the side arm was insulated
to prevent heat loss.

These attempts were partially successful. Whereas the maximum turbine
inlet temperature (the maximum fuel/air ratio) without flashback was
originally only 700°C, the burner could eventually run for about an hour
at a turbine inlet temperature of less than 870°C without flashback. It
is possible that the operational difficulties of the burner are not
difficult to overcome, but the emissions results of this configuration
were disappointing. As the fuel flow increased, the NO_x level also

increased from 62 ppm to 180 ppm. With the burner operating at such a decreased fuel/air ratio ($<870°C$ turbine inlet temperature vs. $1100°C$ desired), the NO_x level should have been less than 50 ppm for the configuration to be attractive.

It is not clear what part of the burner produced the 191 ppm NO_x. Even if the B-4 head end is more prone to generate NO_x than Configurations 1 or 2, it could not have produced this level of NO_x. Unless flashback occurs, there is no combustion in the porous tube. The downstream elements of the burner were the same as those in Configuration 1. Therefore, it appeared that the centripetal swirler was probably responsible for producing NO_x. It is possible that the flashback into the porous tube was an indication of combustion occurring in or directly downstream from the swirler.

It was not considered worthwhile to continue testing Configuration 6 on CDL or residual oil; in order to make Configuration 6 into a viable combustor, the rich burner swirler will have to be more extensively investigated. This could not be accomplished in the screening phase of this project. It was decided to test Configuration 6 as Configuration 13, a hybrid piloted lean burner.

6.5 CONFIGURATION 7 & 8 - RICH PRIMARY CATALYST

Although both configurations were tested, there were no stable runs. During multiple ignition attempts (all rich/lean), the rich catalyst showed remarkable reactivity, but pre-ignition always occurred. The test results and the post-test examinations of test articles indicated that the fuel flow was not brought up fast enough to prevent the catalytic reaction from initiating flashback before the fuel/air ratio entering the catalyst was at the design equivalence ratio.

Catalyst damage also occurred, and it appeared to be primarily mechanical since the α-alumina used for the substrate is quite brittle. It is believed that the failure was due to extreme thermal stresses of stoichiometric conditions on the substrate during the repeated flashbacks.

6.6 CONFIGURATION 9 - MULTIANNULAR SWIRL BURNER

The Multiannular Swirl Burner (MASB) was first tested in the atmospheric test rig to establish ignition characteristics, and subsequently during two test sequences in the pressurized passage. The nominal test conditions during the pressurized tests were as follows:

Table 6-5

NOMINAL TEST CONDITIONS, MASB

Fuels	ERBS, CDL, Residual
Air flow, Kg/sec	2.7
Combustor Inlet Pressure, MPa	1.1
Combustor Inlet Temperature, °C	350
Combustor Outlet Temperature, °C	815 - 1120

Since the primary objective of the project was to select promising low NO_x designs from a large number of candidates, it was not within the scope of this study to test all the possible operating modes of the MASB. In addition, the objective of this study was to demonstrate that the burner is capable of producing a low NO_x level, rather than to explore the modes of operation of the MASB. The NO_x results plotted in Figures 6-8 and 6-9 as functions of ΔT may be interpreted in different ways in terms of the influence of the secondary variables. The selection of the primary zone equivalence ratio (ϕ_p) as the principal parameter is an expedient way of presenting the NO_x results based on the experimental material available at the time of the Phase I report.

The NO_x results with ERBS fuel are plotted against ΔT (Figure 6-8). As ΔT is approximately proportional to the overall fuel/air ratio, the abscissa shows F/A at a different scale. The ΔT values shown were based on the thermodynamically calculated temperature rise rather
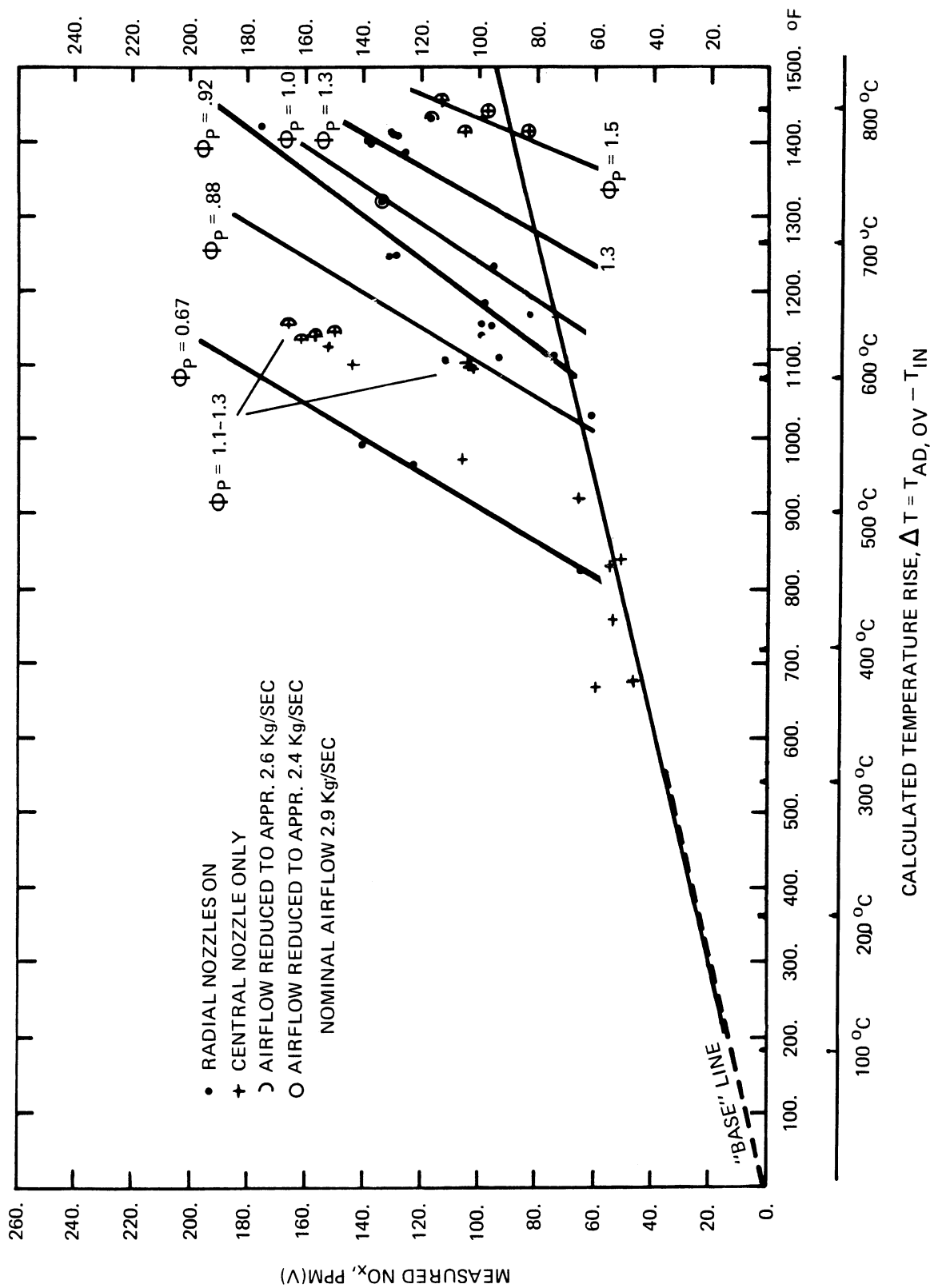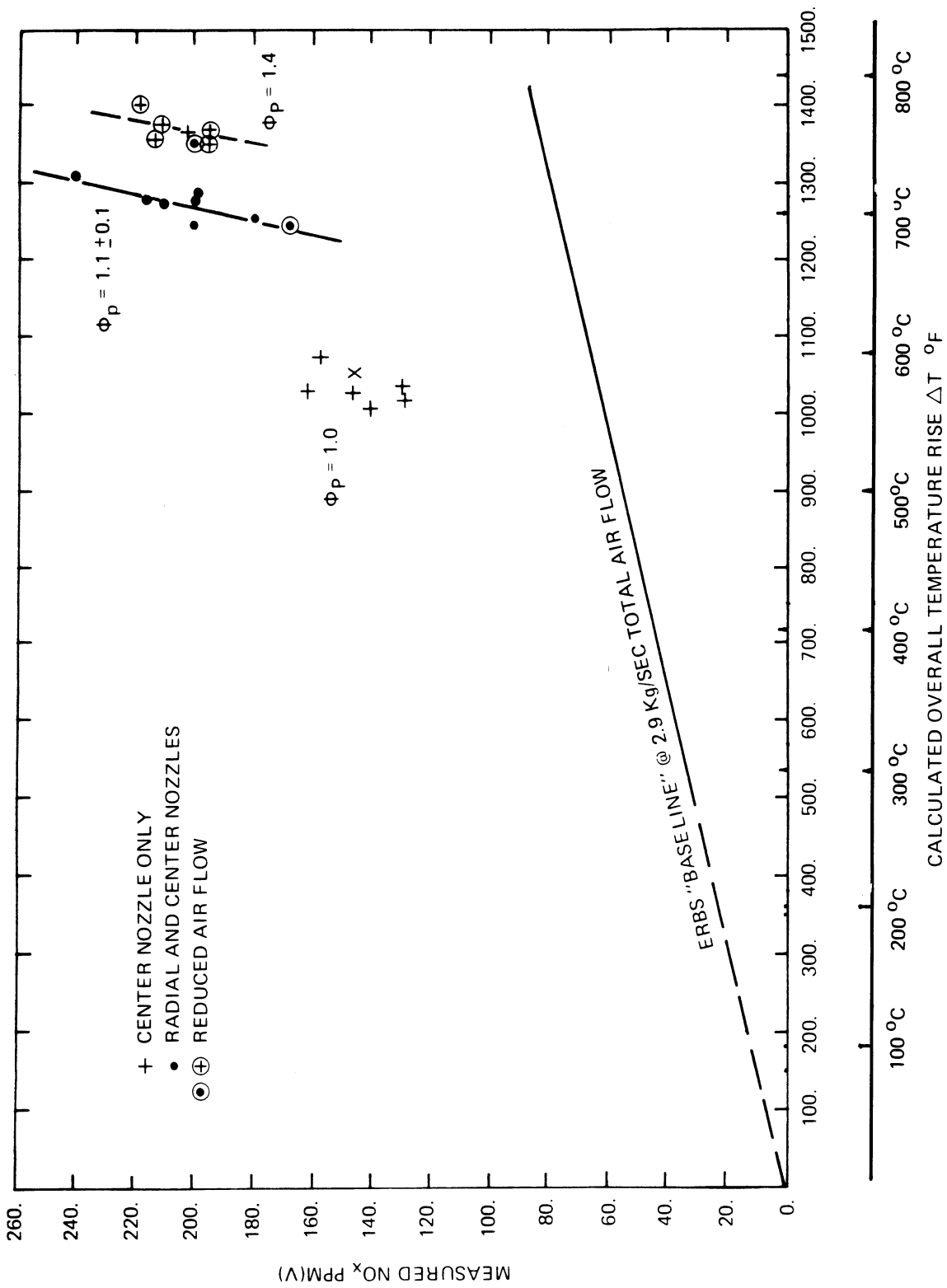
Figure 6-8.   Measured NO_x vs. Calculated Overall Temperature Rise - ERBS Fuel

6-27

Figure 6-9. Measured NO_x Calculated Overall Temperature Rise, CDL Fuel

than on the average gas temperature measured downstream of the dilution air (burner outlet).

NO$_x$ RESULTS ERBS FUEL

The presentation of the test data (Figure 6-8) is based on the following assumptions:

- For a given air flow, NOx is a linear function of the fuel/air ratio up to a certain maximum value of the F/A

- When this maximum fuel/air ratio is reached, the increase in NOx as a function of the fuel/air ratio again follows a quasi-linear relationship, but the slope is much greater

- The change in the slope can be delayed to higher F/A values, provided the head end (primary zone) of the burner is made fuel/rich (ϕ_p is increased)

Most of the test points were obtained at a 2.9 Kg/sec total air flow (nominal conditions). The exceptions (2.6 and 2.4 Kg/sec) are noted in the figure. Test points with crosses indicate that the radial nozzles were also in operation.

The assumptions listed above may be illustrated with the test data. During some test runs, the head end equivalence ratio, ϕ_p, was 0.67 (nominal); that is, the innermost three swirlers and the fuel flow from the central nozzle produced a lean mixture in the primary zone. The slope change occurred at $\Delta T \sim 415°C$. By increasing the head end equivalence ratio to $\phi_p = 0.88$, the slope change could be delayed to 565°C. These two $\phi_p = $ const. lines are fairly well established, especially the $\phi_p = 0.92$ line, which is based on eight test points. The other ϕ_p lines ($\phi_p = 0.88$, 1.3) were drawn through test points taken within a narrow ΔT range. However, it was assumed that their course will be similar to the $\phi_p = 0.92$ and $\phi_p = 0.67$ lines. If $\phi_p = 1.5$ was obtained, the slope increase will be further delayed to $\Delta T = 730°C$ either by using both the central and the radial nozzles, or by using the central nozzle alone.

Trying to make the head end fuel-rich by reducing the air flow in the primary zone, rather than by increasing the fuel flow may not delay the increase in slope (test points ϕ_p 1.1 - 1.7). It is not clear why the same ϕ_p = 1.4 value is capable of delaying the slope change when it is obtained by an increase in the fuel flow whereas a slope change occurs when ϕ_p = 1.5 is obtained by decreasing the primary air. It is possible that because a decrease in the primary air not only makes the head end more fuel-rich, but also increases the residence time in the head end, the thermal NO_x generated in the head end will increase.

NO_x RESULTS, CDL AND RESIDUAL FUEL

The few tests run with coal-derived liquid (Figure 6-9) demonstrate that the MASB is capable of producing low NO_x levels comparable to those of a modular burner such as Configuration 1. The lines were drawn to suggest that the full map of the burner operation with CDL fuel might show similar characteristics as the ERBS operation (Figure 6-8). The FBN conversion rate in one test run was approximately 20 percent, which indicates that the MASB is a rich/lean burner. The single operational point with residual oil indicated a NO_x level of 120 ppm at ΔT = 725°C. This NO_x level corresponds to approximately 27 percent conversion of FBN. The increase from 20 percent with CDL can be expected because of the lower FBN level of the residual oil (0.22 percent FBN instead of the 0.8 percent FBN in the CDL).

NO_x RESULTS, FUEL MIXTURES

During the transfer from ERBS to CDL, a ΔNO_x of approximately 97 ppm was observed between 100 percent ERBS and 100 percent CDL operation. NO_x measurements at 26 percent, 51 percent and 77 percent CDL fuel flow indicated 141, 143 and 172 ppm NO_x, respectively. These values correspond to 43, 45 and 75 ppm ΔNO_x. The 45 and 75 ppm ΔNO_x values are approximately the right levels for the mixture; the 43 ppm at 26 percent CDL is higher than the approximate 25 ppm ΔNO_x expected. It is possible that the NO_x sample was not taken at the corresponding flow split.

A 73 percent residual/ERBS mixture was also sampled for NO_x during the transition to residual oil. The NO_x was 123 to 137 ppm, corresponding to 25 to 39 ppm if 98 ppm is considered to be the ERBS base NO_x at the operating conditions of the fuel transfer. Both values are higher than the NO_x at 100 percent resid (120 ppm). It is believed that the mode of operation or the timing of the sample taking was not adequate, and the resid operation may not have been run at the ERBS operational point where NO_x was 98 ppm.

NORMALIZATION OF THE MASB TEST DATA

It appears (Figure 6-8) that the most important aspect of the NO_x performance of the MASB on ERBS fuel is defined by the baseline, which is the minimum NO_x level for the geometry tested. Departures from this line may be avoided if the head end equivalence ratio (ϕ_p) is increased as F/A (or ΔT) increases. Other interpretations of the data are possible, but more investigation is necessary to describe the process definitively. The test points that constitute the base line were replotted in Figure 6-10 using the NO_x values corrected to 15 percent O_2. If the $(NO_x)_{meas}$ vs. ΔT plot is linear and goes through the origin, the NO_x corrected to 15 percent O_2 vs. ΔT plot should be a constant. Based on the curves (Figure 6-8), this condition is expected.

When the correction was first made using the measured NO_x and O_2 values, excessive data spread was observed. The corrected NO_x values did not adequately define a constant $(NO_x)_{correct}$ value. The difficulty was traced to a data spread of the oxygen readings. Therefore, the observed oxygen readings were substituted with oxygen data calculated from the measured fuel/air ratio. The NO_x corrected to 15 percent O_2 for the base line (Figure 6-10) is approximately 84 ppm.

The baseline discussed above may also be considered as a load vs. NO_x curve because the turbine inlet temperature (T_{IT}) acts as the principal variable when the load changes.
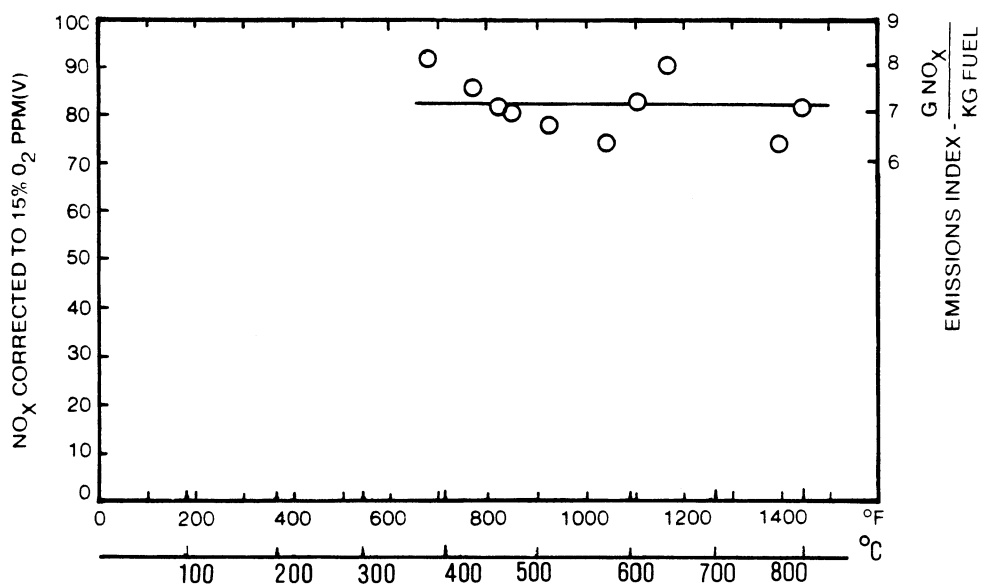
Figure 6-10. "Baseline" NO_x corrected to 15% O_2 vs. calculated overall temperature rise

Pressure drop in the MASB was favorable: it was two to four percent depending on the overall air flow rate.

The smoke level in the MASB at the lowest NO_x level was 41 SAE number. Further investigations will be necessary to reduce this smoke level. No attempt was made to influence the smoke level by changing the atomizing air flow, which is the most obvious variable to change in order to reduce smoke. Table 6-6 shows the smoke results of the MASB for the characteristic tests selected. The table also shows the combustion efficiency-based exhaust gas analysis, and the CO and unburnt hydrocarbon levels that were input for the efficiency calculation.

Table 6-6
EMISSIONS LEVELS, MASB

Fuel	SAE Smoke	ϕ_p	% η_{comb}	UHC @ 15% O_2 ppm	CO
ERBS	34	0.83	98.8	17.9	421
ERBS	34	0.89	99.0	103.1	665
ERBS	11	0.93	99.5	17.9	421
ERBS	15	0.94	99.5	19.7	499
ERBS	43	1.48	99.9	14.8	40.7
ERBS	41	1.47	96.3	10.8	96.3
CDL	2	1.38	99.9	8.8	47
CDL	Very Low	0.99	99.9	8.09	110.4
CDL	18	1.02	99.8	10.9	169.1

It can be seen that the combustion efficiency is uniformly high in all the cases investigated; the unburnt hydrocarbon level is negligible and, except in Run #2, the CO is also acceptable.

MASB SUMMARY

The multiannular swirl burner is a low emission, low pressure drop burner design; it can meet EPA NO_x requirements dry and, running on CDL at 0.8 percent FBN, the burner almost meets EPA requirements.

6.7 CONFIGURATION 10 - PERFORATED PLATE, VENTURI QUENCH

This configuration was tested with two different fuel preparation zones and with two perforated plates with different open area dimensions. During multiple ignition attempts, both lean and rich/lean, either pre-ignition or an unstable flame was observed before a steady state was achieved. In several attempts, flashback was observed at conditions in which it does not occur with catalytic combustors. Had the perforated plate been cooled to prevent upstream surface heating, the flashback may have been prevented.

6.8 CONFIGURATION 11 - ROLLS-ROYCE

As detailed in Section 3, this combustor was modified after the initial
tests and retested. The following presentation of results includes both
combustors; the modified combustor will be referred to as the richened
primary zone (RPZ) combustor. Test results are presented in Figures 6-11
through 6-13.

THERMAL NO_x RESULTS

The ERBS results in Figures 6-11 and 6-12 show that Configuration 11 is
a low thermal NO_x producer. At the design point a value of 87 ppm
corrected to 15 percent O_2 was obtained.

The extent of primary zone richening did not appear to strongly affect
thermal NO_x production. The measured data (Figure 6-11) suggests a
tendency for thermal NO_x to be lower in the RPZ combustor at conditions
richer than design point. This trend does not appear when the results
are corrected to 15 percent O_2. For oxygen-corrected results and emis-
sion indices it is not possible to discern separate trends for the RPZ
combustor.

FUEL BOUND NO_x RESULTS

Figure 6-13, FBN conversion versus overall ϕ, shows a reducing trend in
conversion with richening (increasing ϕ) for both residual and CDL fuels.
Rig fuel flow limitations inhibited the full design ϕ; the maximum values
shown correspond to primary zone recirculation ϕs of about 1.15 for CDL
and 1.4 for residual fuel.

Throughout the results, there is a marked reduction in total NO_x produc-
tion with the modified combustor. As this reduction was not apparent in
thermal NO_x, it must therefore signify a reduction in FBN conversion.
This deduction is supported by the RPZ data (Figure 6-13). Residual
fuel results with the RPZ combustor are limited to two points only; for
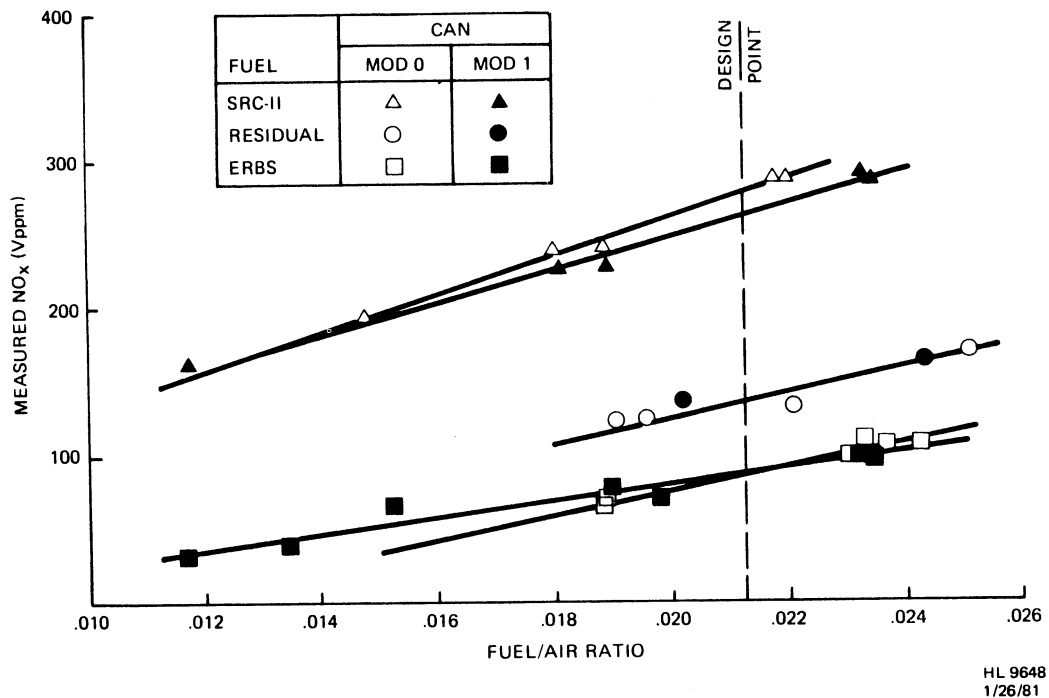
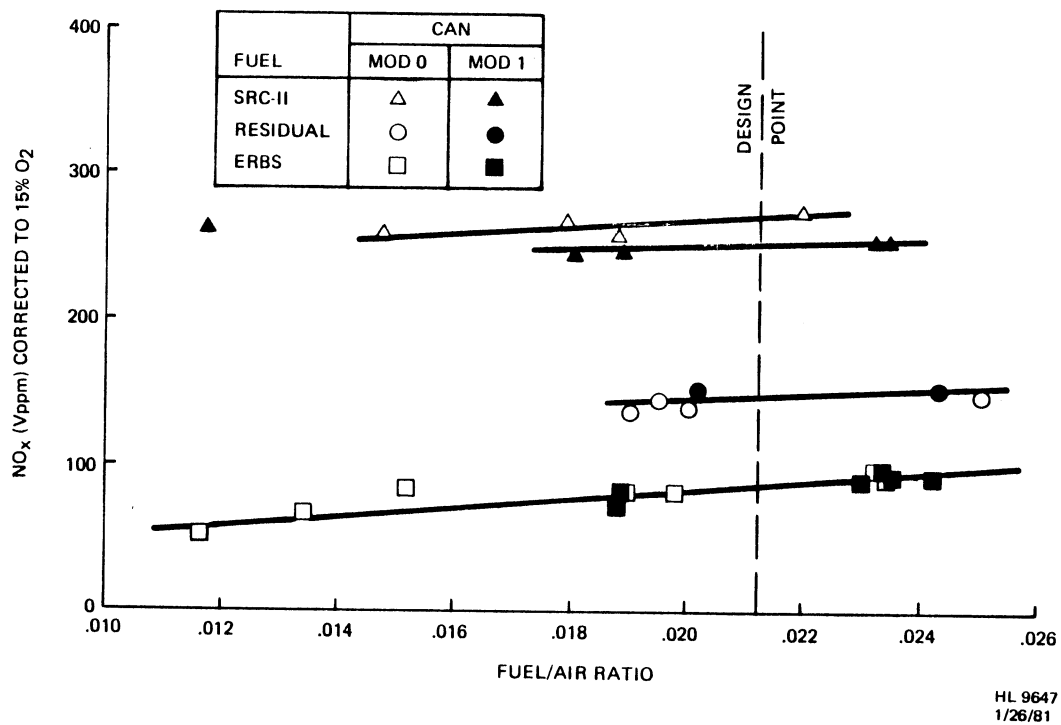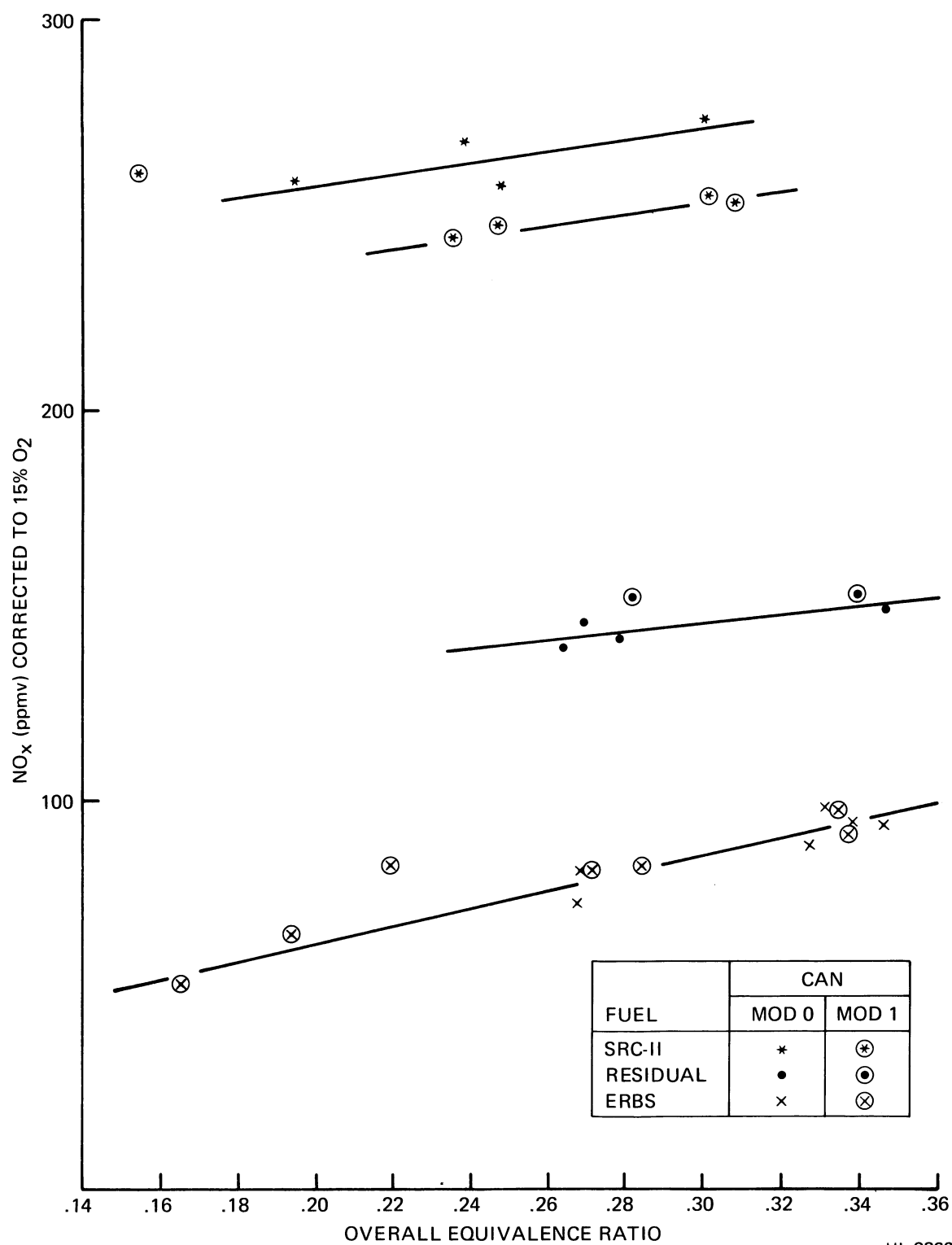Figure 6-11. Measured NOx (ppmv) vs. Fuel/Air Ratio Rolls-Royce Combustor

Figure 6-12. NOx (ppmv) Corrected to 15% O_2 vs. Fuel/Air Ratio – Rolls-Royce Combustor

Figure 6-13.   NOx (ppmv) Corrected to 15% O_2 vs. Overall Equivalence Ratio

this reason, the apparently different trend in this data is not particularly significant.

COMBUSTION EFFICIENCY

Emissions of carbon monoxide and unburnt hydrocarbons are extremely low for all fuels in both combustor configurations. The unburnt fuel results show a steep increase for the RPZ combustor, but combustion efficiencies are consistently close to 100 percent for all power conditions and are very high (98.4 percent +) for simulated gas generator idle.

COMBUSTOR PRESSURE DROP ΔP PERCENT

Combustor pressure drop varied from 6 to 12 percent in the load range of interest. It is believed that the higher readings are in error.

SMOKE

Throughout the test series, there were problems with the smoke sample transport and measuring system. The little available data for the unmodified combustor shows smoke to be consistently very low on all three fuels. Although the data base is very limited, acceptable levels of smoke below the visible threshold can be expected. Visual observations of the RPZ exhaust plume suggested no serious deterioration in the unmodified combustor.

OUTLET TEMPERATURE VARIATION

Outlet pattern factor (PF) is consistently very low - below 3 percent for all conditions. The remoteness of the measurement plane downstream from the combustor outlet contributes to this low PF.

FLAME TUBE METAL TEMPERATURES

Flame tube metal temperatures were not measured directly by either

thermocouple or thermal paint. The combustor was inspected after each run and there was no evidence of overheating during normal running.

SUMMARY OF CONFIGURATION 11

The original high mix combustor has proven to be an efficient, smoke-free, mechanically reliable device capable of low NO_x emissions with non-nitrogen containing fuel; it is also tolerant of a wide range of fuel qualities. The achieved NO_x level, 87 ppm corrected to 15 percent O_2 at peak, will meet the EPA requirement of 75 ppm when corrected for humidity and operated in a system with typical cycle efficiency (31 percent).

Performance on nitrogen-bearing fuels was, with regard to FBN conversion, not markedly better than a conventional combustor. The best value achieved on CDL fuel was 52 percent FBN conversion. Modifications to richen the primary zone recirculation to $\phi = 1.6$ (although this level was not fully achieved) did show a reduction in FBN conversion without affecting smoke production or combustion efficiency.

The attempt to richen the primary zone was not fully successful because the existing primary zone arrangement is aerodynamically dominated by the swirler cuff. This feature constrains the swirler efflux from flowing outward down the combustor head and walls, and centralizes it. These conditions produce a rich central reaction zone with a port flow that is drawn back close to the wall before it enters the reaction. Thus, there is very good performance on residual fuel. By blanking the swirler, the flow through it into the reaction was reduced; but as the powerful cuff dominated, recirculation was starved of air and more air was entrained from the secondary port flow. To substantially richen the primary zone, the interdependent cuff/port flow effect must be prevented.

The best way to prevent FBN conversion is to richen the primary re-circulation, increase the residence time within the rich zone, and maximize inlet temperature. As inlet temperature is predetermined in this exercise, the following recommendations for combustor modification are proposed.

RECOMMENDATIONS FOR IMPROVEMENT OF CONFIGURATION 11

To break the dominant cuff/port relationship, the primary zone may be lengthened by moving the first row of ports downstream. This will reduce the proportion of port flow involved in the primary zone recirculation, consequently richen the reaction, and proportionally increase primary zone residence time.

Although it is difficult to determine a new primary zone length, it will depend on how a change in length affects FBN conversion and smoke performance. For convenience, the first row of ports should be removed and inserted downstream of the quench zone, thus increasing the primary zone length.

The expected effect of this modification will be a drastic reduction in FBN conversion and a slight reduction thermal NO_x by reducing the primary zone temperature. Other Rolls-Royce evidence suggests that a 10 percent thermal NO_x reduction may be expected. The ability to use heavy fuel is likely to be unaffected, and no severe mechanical problems are expected. Smoke production will probably increase, but as the most recent tests have shown, the existing combustor is remarkably smoke-free so this increase can be tolerated.

6.9 CONFIGURATION 12 - LEAN CATALYTIC

This configuration uses a lean combustion catalyst and a premixing, prevaporizing fuel preparation system. The catalytic combustor was generally run at an inlet pressure of 1.2 MPa and an inlet temperature of 390°C in the lean mode; the fuel/air ratio was 0.022 and the reference velocity, 23 M/S. The data for each fuel tested is explained below.

ERBS FUEL RESULTS

Test runs on ERBS fuel produced NO_x emissions of less than 1 ppm with combustion efficiencies greater than 99 percent. One of the test program goals was to demonstrate a configuration capable of ultra-low NO_x emissions, defined as less than 50 percent of the Federal EPA limit, when testing on clean fuels. The catalytic combustor operating in the lean mode achieved this goal in every test with ERBS fuel.

ERBS Pyridine-Doped Results

Tests were also run with ERBS fuel doped with pyridine to determine the influence of nitrogen concentration in the fuel on NO_x concentration. Figure 6-14 shows NO_x concentrations in the exhaust corrected to a constant dilution (15 percent O_2) as a function of the nitrogen concentration in the fuel. The result is nearly linear. Nitrogen concentration in the conversion of FBN to NO_x is shown in Figure 6-15. It is clear that the conversion level is only moderately influenced by the FBN, and that it is approximately 50 percent or less. This result was totally unexpected since other results from lean catalytic combustors usually exhibit a high conversion of FBN to NO_x.
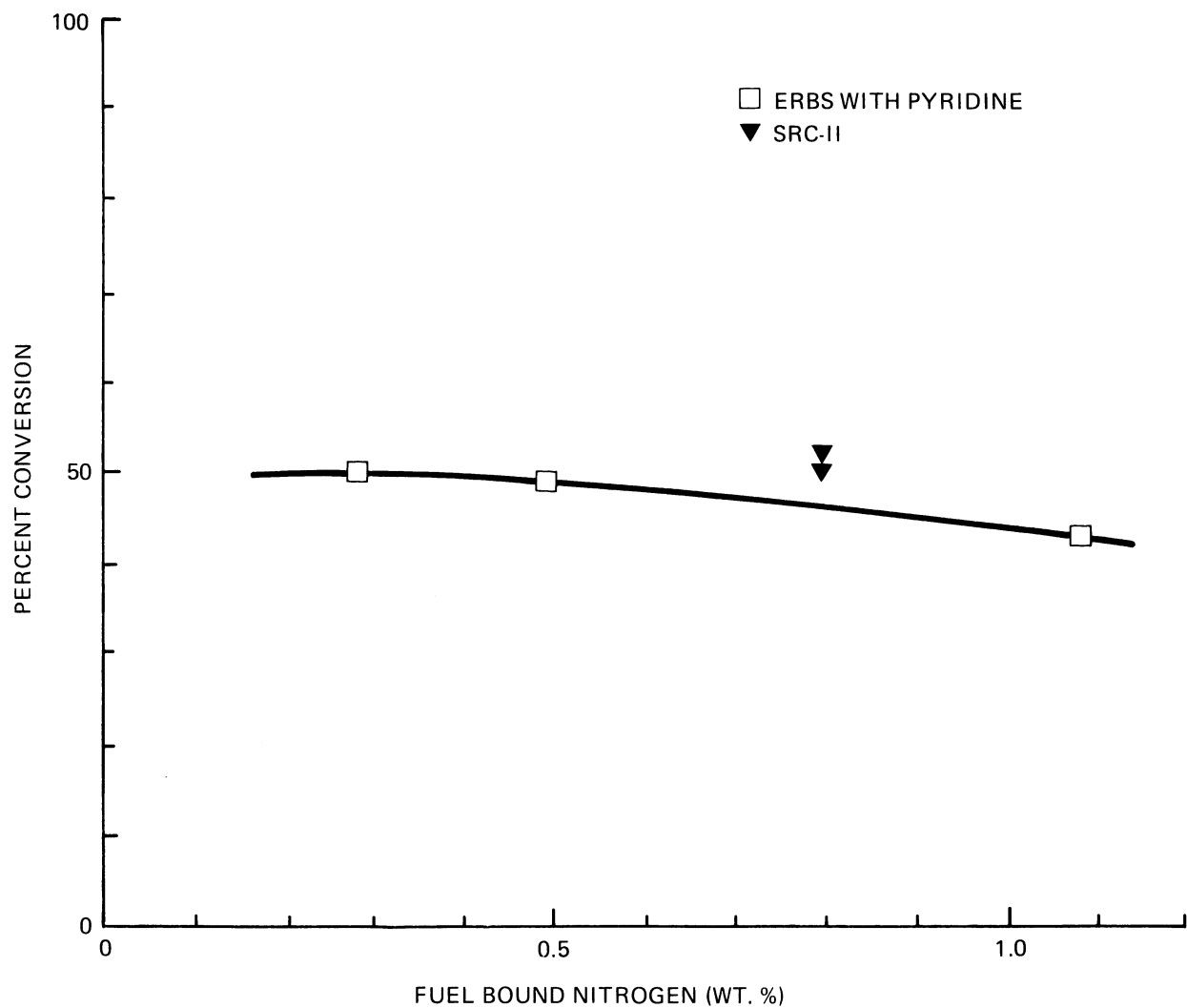
SRC-II RESULTS

Tests were also run using SRC-II fuel. (NO_x concentrations obtained are included in the plots of Figures 6-14 and 6-15.)
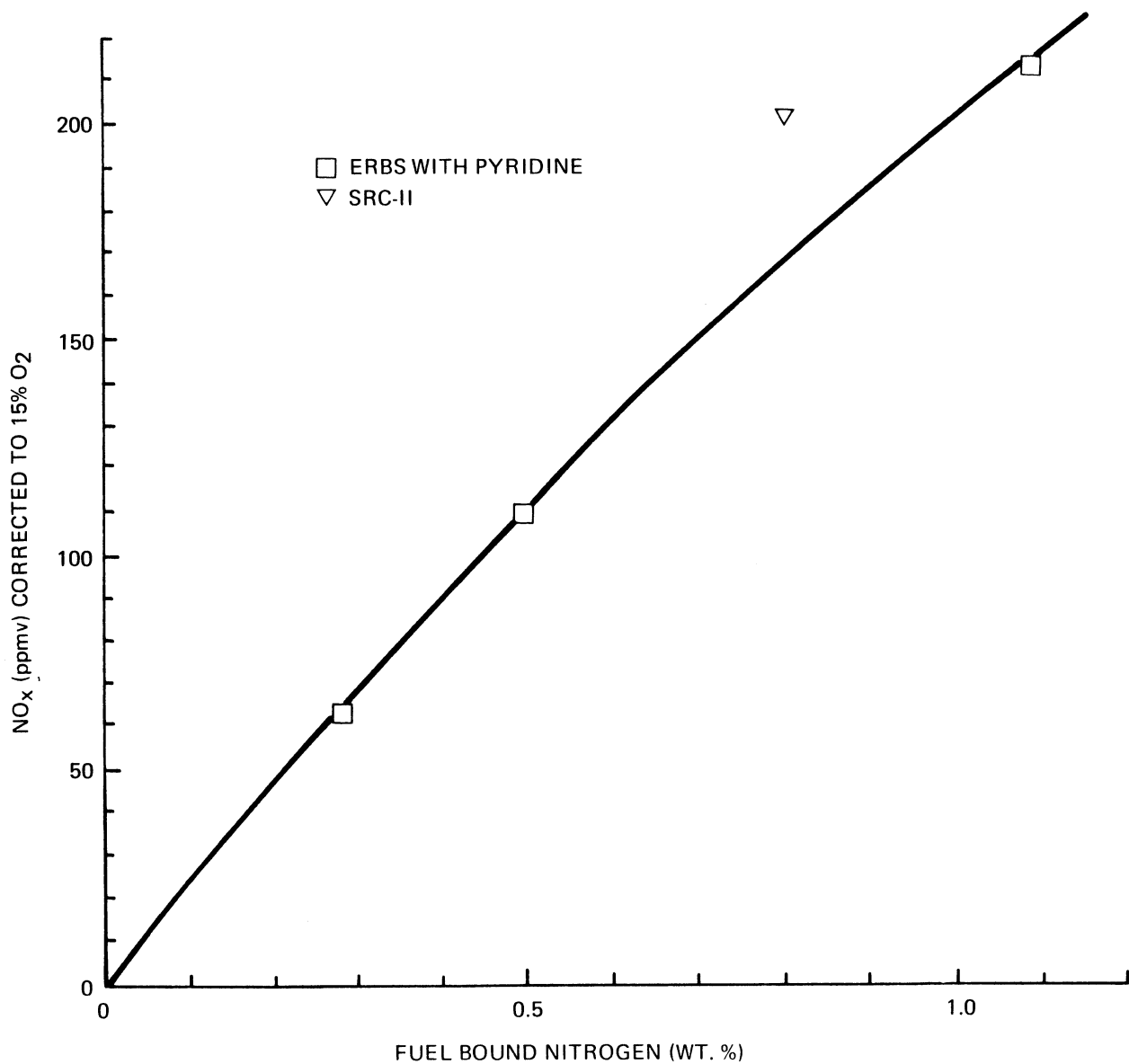
Figure 6-14. NO_x Corrected to 15% O_2 Vs. Fuel Bound Nitrogen for Lean
Premixed Prevaporized Catalytic Combustor

Figure 6-15.   Conversion of Fuel Bound Nitrogen to NO_x Vs. Fuel Bound
Nitrogen for Lean Premixed Prevaporized Catalytic Combustor

The results of SRC-II support the results of the pyridine addition regarding FBN conversion to NO_x. The test runs using SRC-II did not exactly replicate the condition used for the pyridine runs, but the results are consistent.

The SRC-II runs were characterized by an easier light-off than the ERBS runs. Apparently, the SRC-II was a more easily oxidized fuel than ERBS in the catalyst. There was one drawback with SRC-II fuel in Configuration 12: it was prone to occasional pre-ignition in the fuel preparation zone. The fuel did not cause a deposition or fouling problem either on the catalyst or on the fuel injector.

Certain SRC-II runs showed a relatively high amount of the NO_x as NO_2: 28 of the NO_x as NO_2 was a typical value. Since it did not always appear with the SRC-II, this result is apparently unique to SRC-II only at certain operating conditions. It was never observed when testing with other fuels.

SUMMARY OF CONFIGURATION 12

The most significant findings of the testing of Configuration 12 are summarized below:

- Extremely low NO_x emissions were measured on clean fuels

- Conversion of FBN to NOx was approximately constant for all FBN concentrations

- Approximately 50 percent of the FBN was converted to NO_x

6.10   CONFIGURATION 13 - HYBRID PILOTED LEAN BURNER

Configuration 13 has the same hardware as Configuration 6. The difference is that the premixed side arm is not run as a fuel-rich, but as a premixed lean burner.

In order to run the burner in this fashion, the ratio between primary combustion air and quench air (secondary air) was changed from 0.3 Kg/sec to 0.93 Kg/sec in Configuration 6, and from 0.5 Kg/sec to 0.23 Kg/sec in Configuration 13. A lean steady state point was obtained: NO_x was 75 ppm at $T_{IT} < 815°C$.

Due to local overheating in the quench section (resulting from low quench flow), shutdown was required after this steady state run.

Post-test inspection of this geometry showed massive damage to the swirler component of the vortex combustor. Approximately 90 percent of the material melted, and it was blown into the quench and lean burner sections.

There was no indication that the damage occurred during the lean operation. This condition agrees with the possibility mentioned in Section 6.4: the large centripetal swirler acted as a high temperature lean burner instead of a rich mixer. The conclusion, therefore, is the same as in Section 6.4: to further develop Configuration 13, the swirler burner requires further study.

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1. Sarofim, A. F., "Combustion Chemistry and Modeling: An Overview," Proceedings of Symposium on Stationary Source Combustion, EPA, Sept. 1975.

2. P. Walsh, et al, "Formation & Control of Fuel-Nitrogen Pollutants in Catalytic Combustion of Coal-Derived Gases," Final Report, DOE Contract No. EF-77-S-01-2762 (1 Feb., 1980).

3. R. V. Carrubba, et al, "Method of Combusting Nitrogen-Containing Fuels," U.S. Patent 4,054,407, (18 Oct., 1977).

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Section 7
CONCLUSIONS FROM TEST RESULTS AND RATING OF COMBUSTORS

The work plan for this project focused on 13 configurations. All have been built, and all but two have been tested. Table 7-1 summarizes the results of the tests.

Configurations 3 and 5 were not tested. They were eliminated from the program because components other than the perforated plate and the recirculating swirl yielded adequate results. Configuration 4 (tested) performed adequately, but not better than Configuration 2, which is simpler. Other configurations had serious operational problems and their performance did not warrant the effort to solve these problems (Configurations 6, 7/8, 10 and 13). This left Configurations 1, 2, 9, 11 and 12 for further consideration. All five had noteworthy performance features.

Configurations 1 and 2 are similar in that both are modular designs using the same components (except the quench section). The main difference in performance between the two was that Configuration 2 had a 50 ppm higher NO_x output on ERBS fuel. As explained in Section 3.2, the difference may originate from the effects of the swirl vortex quench module used in Configuration 2. For the purposes of this evaluation, Configurations 1 and 2 will be considered as a single concept, the "modular combustor". The choices between using a vortex or venturi type opposing jet quench module will be made on the basis of future development studies. The following evaluation therefore focuses on four configurations:

Table 7-1

SUMMARY OF TEST RESULTS

Configuration		Tested	Remarks
1	Direct Injection, Opposing Jet Venturi Quench, Straight Pipe Lean	Yes	Low NOx for ERBS, low FBN conversion. Acceptable smoke, highest pressure drop
2	Direct Injection, Vortex Quench, Straight Pipe Lean	Yes	Moderately low NOx for ERBS, low FBN conversion, high smoke
3	Direct Injection, Vortex Quench, Perforated Plate Lean	No	Straight pipe lean is adequate
4	Direct Injection, Vortex Quench, Catalytic Lean	Yes	Not better than straight pipe lean
5	Recirculating Counter Swirl, Opposing Jet Venturi Quench	No	Direct injection rich burner is adequate
6	Hybrid Piloted Rich Burner Opposing Jet Venturi Quench	Yes	Both operational and performance problems were encountered
7/8	Catalytic Rich, Opposing Jet Venturi Quench, Straight Pipe Lean	Yes	Difficulties in operating the rich catalyst
9	Multiannular Swirl Burner	Yes	Acceptable NOx for ERBS, low FBN conversion; lowest pressure drop
10	Perforated Plate Rich Burner Opposing Jet Venturi Quench	Yes	Direct injection rich burner is adequate; also, operational problems encountered
11	Rolls-Royce	Yes	Acce table for ERBS but high FBN conversion
12	Lean Catalytic	Yes	Excellent for ERBS high FBN conversion
13	Lean Hybrid Premix/Direct Injection	Yes*	Both operational and performance problems were encountered using Configuration 6 hardware.

* Run with Configuration 6 Hardware.

o   Modular Rich/Lean

o   Integral (Rolls-Royce)

o   Multiannular Swirl

o   Catalytic (Lean)

7.1 CHARACTERIZATION OF SELECTED CONFIGURATIONS

NOx Performance

All four configurations met the EPA limit on ERBS fuel, and the lean
catalytic burner met the program goal of ultra-low NO_x emissions
on ERBS fuel. When operated on fuels with bound nitrogen, the modular
rich/lean and multiannular swirl burners have much lower conversion
efficiencies and are therefore more favorable than the Rolls Royce and
catalytic burners, depending on the FBN level. All four burners can be
optimized to further reduce NO_x.

Smoke

Smoke levels in all four burners met or were close to the program goal
of SAE 20. In some cases, this conclusion is based on actual observation
of the exhaust. No attempt was made in this screening program to reduce
smoke, as it was felt that this could best be accomplished in a full
scale burner.

CO and UHC

Carbon monoxide and unburnt hydrocarbon levels were not high enough in
any of the burners to cause any concern. These results are consistent
with the high combustion efficiencies (99+ percent) obtained for all
burners.

Structural Materials

The multiannular swirl burner and the Rolls Royce burner are of conven-
tional metallic sheet construction, and no durability or maintainability
problems are expected.

The rich/lean and catalytic burners depend on ceramic materials to achieve low NO_x performance, and materials development programs will have to parallel any effort to scale these burners to utility sized applications.

Pressure Drop

The multiannular swirl and Rolls Royce burners have lower pressure drops than the other two; they are therefore preferred for actual application where high cycle efficiency is required.

Section 8
FULL-SCALE DESIGN

8.1 FULL-SCALE DESIGN, MULTI-ANNULAR SWIRL BURNER (MASB)

This section describes the philosophy and processes for scaling the laboratory-size burners to a full-scale industrial combustion turbine. The multiannular swirl burner was selected as an example.

Scaling is the most important factor in developing a full-size burner from a laboratory model. Experience in the combustion turbine industry has shown that keeping pressure, temperature, reference velocity and fuel/air ratio in the small-scale burner equal to the conditions in the full-size combustor will produce full-size combustion performance similar to the scaled version. Comparable combustion performance can be attained, although equal reference velocity means longer residence time in the full-scale burner.

The production of a full-scale, low emissions burner, however, presents additional scaling considerations. The basic combustion reactions in both burners are fast enough to be completed within the time allowed by reasonable reference velocities. However, both the small and the full-scale combustor emission-forming reactions (thermal NO_x for example) are much slower, and may proceed further in the full-size burner than in the small burner. Because of a longer residence time in the full-size burner, it may produce more thermal NO_x than the small burner.

The rich/lean burners in the present program, however, are not conventional. They were deliberately constructed to interfere with NO_x

formation. Results obtained during the first phase of the program indicated that at least some NO_x formation in a rich/lean burner proceeds along reaction sequences that are different from the Zeldovich reactions considered in the formation of thermal NO_x in a conventional combustor. During the testing of the MASB, for example, a 20 percent increase in the residence time (caused by a 20 percent decrease of the mass flow) caused a 25 percent decrease in thermal NO_x. Consequently, the full-scale MASB will be designed for the same reference velocity (longer residence time) as the small-scale burner.

In addition, the reaction time of the release of fuel-bound nitrogen from the fuel is comparable to that of the combustion reactions. It was possible to achieve FBN conversion levels in the MASB comparable to those in the modular, rich head end burner (Configuration 1) where the residence time was much longer than in the MASB. It was therefore decided to design the full-scale MASB with a geometry similar to the scaled burner and to keep the reference velocity equal to that of the small scale burner.

AERODYNAMICS

Radial swirl distribution will be the same as in the small-scale version shown in Figure 8-1. The innermost annulus is 60° (air angle), reduced gradually to 20° in the outermost annulus. This arrangement simulates a free vortex. The central tube will have a small controlled air flow around the nozzle; therefore, the tube ID is somewhat larger than the nozzle diameter. The small air flow through the tube will be controlled by throttling from the outside (upstream of the central tube).

As in the small burner, the full-scale burner will have five annuli. The relative (radial) width of the annuli should be the same as in the scaled burner, although this may be modified if emissions and/or pattern factor considerations warrant a somewhat different radial air distribution.
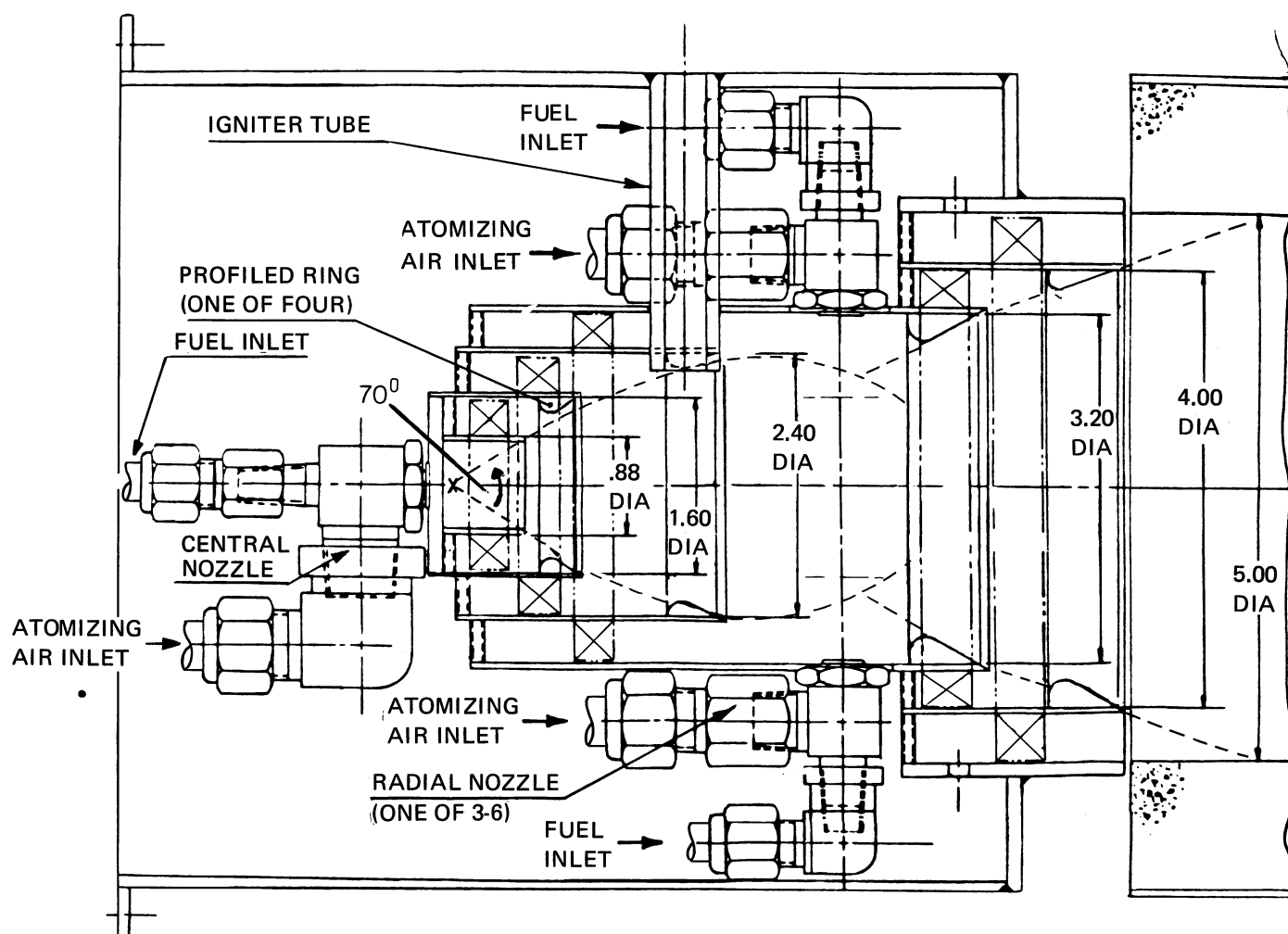
Figure 8-1.   Multiannular Swirl Burner (MASB)

Axial Flow Field

Design of the axial flow field will be determined by:

- Relative distance of the annuli
- Location of the secondary fuel injection nozzles

Axial spacing will be satisfactory when the flame fills out the contours provided by the annuli and their profiled ring inserts (Figure 8-1). In the design of the scaled version, it was assumed that a recirculation zone developed as a result of the collapse of the swirl exiting from the innermost two annuli. Qualitatively, this was expected from the original atmospheric version of the burner. The successful performance of the pressurized version indicates a satisfactory flow field.

In the scaled pressurized tests, a fairly large increase in the residence time due to a reduction of the air flow decreased the NO_x output instead of increasing it as expected from experience with conventional burners. Consequently, if the plug flow velocity in the full-size burner is the same as in the scaled version (which means an increase in the residence time compared with the small burner) there may not be an increase in NO_x in the full-scale burner.

It appears that geometric similarity may be retained without raising the NO_x level. Before building the full-size burner, however, pressure tests at 6 atm and 15 atm should be run on the small-scale burner to verify this thesis. These tests examine the influence of residence time without changing the fuel/air ratios. The influence of the changed pressure on thermal NO_x alone may be deduced from the square root relationship between thermal NO_x and pressure. The influence of residence time can then be separated from the results.

For the purposes of this discussion, it is assumed that the increase in residence time calculated on a plug-flow basis will not increase NO_x

production; that is, the results of the tests suggested above will be anticipated.

CALCULATION OF THE SCALING FACTOR

The calculation of the scaling factor between the small test combustor and a full-size burner is based on the parameters of an advanced technology, 100-MW-class turbine designed specifically for high reliability. The most relevant parameters are shown in Table 8-1.

Table 8-1

DESIGN PARAMETERS, FULL SIZE

Output MW	104.17
RPM	3600
Generator efficiency, %	98.5
Heat rate, Btu/Kwh	10430
Compressor inlet airflow, Kg/sec	351.8
Number of combustors	14

	Full-Scale	Small Scale (5.0 In. Dia.)
Air flow/combustor, Kg/sec	20.4	2.4
Shell pressure, MPa	1.4	1.13
Turbine inlet pressure, MPa	1.34	1.11
Shell temperature, °C	372	345

The basis of scaling is the reference velocity, V_{REF}:

$$(V_{REF})_S = (V_{REF})_F.$$

The reference velocity of the full-sized combustor $(V_{REF})_F$ should be the same as that of the scaled version $(V_{REF})_S$. The customary definition of the reference velocity is based on the volume flow under shell conditions, but the characteristic cross section is at the combustor exit just

8-5

before the flow enters the transition piece. Usually, this section is the largest diameter of the combustor; in the scaled burner the characteristic diameter is 0.127M (Figure 8-1). Using the values shown (Table 8-1), the reference velocity is calculated to be 29.4 M/sec.

This reference velocity is about 20 percent lower than the reference velocity in conventional burners. Note that the MASB was designed for 2.9 Kg/sec air flow, which produced a reference velocity of 36.1 M/sec. However, it was found that reducing the mass flow from 2.9 Kg/sec to 2.4 Kg/sec considerably decreased the NO_x output level. Therefore, the value 29.4 M/sec will be used for reference velocity in case full-scale tests indicate that acceptable NO_x levels can be achieved at higher V_{REF} (e.g., by changing the primary zone equivalence ratio). It will thus be possible to process a higher mass flow through one burner as the pressure drop in the scaled combustor was only two to three percent.

Using the value V_{REF} = 29.4 M/sec, the characteristic diameter of the full-size burner is calculated to be 0.34M.

Since the small-scale burner (Figure 8-1) diameter is D_S = 0.12M, the scaling factor of the MASB is

$$S_F = \frac{0.340}{0.127} = 2.68$$

FULL-SCALE MASB

Using the factor S_F, the key dimensions of the burner shown (Figure 8-1) were enlarged and a design was developed for the 104 MW turbine as shown in Figure 8-2. These dimensions were:

- Mean diameter of annular swirlers
- Radial dimension (annular width) of swirlers
- Axial distance, swirler exit to swirler exit
- Axial location of the radial nozzles
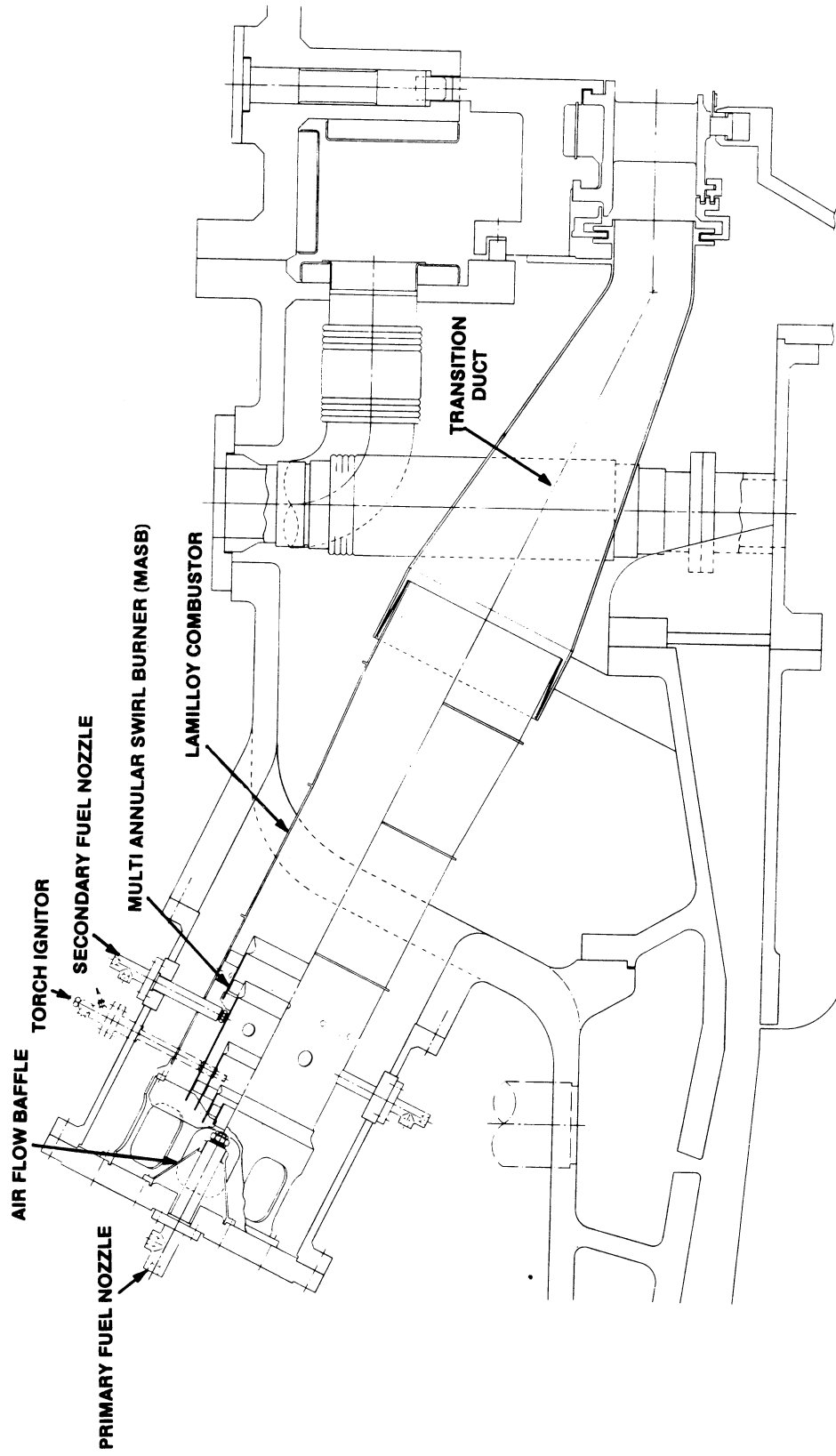- Profile of rings in the swirler exits

Figure 8-2. Full Size Multiannular Swirl Burner (MASB) (Source of Data: EPRI RP1187-2 Phase II Westinghouse HI-REL Program)

Swirler exit is defined as the trailing edge of the profiled rings, not the exit cross section of the swirler vanes. Therefore, the axial location of the vane rows within their annular space is not necessarily the same in the scaled (Figure 8-1) and full-size burner (Figure 8-2). For example, locating the igniter in the turbine shell dictated that the third annular row of vanes be moved upstream in the full size burner. It was assumed that once the swirl was located in the annular space, it would proceed in the annular space without substantial change until it arrived at the discharge end (trailing edge) of the annular swirler. The relevant scaling relates to the free space between the profiled rings where the combustion reaction takes place in the swirling flow.

Another important feature of scaling is the arrangement of the central nozzle with respect to the cylinder that carries the innermost swirler. Although the central nozzle is shown in an extreme upstream position (Figures 8-1 and 8-2), the actual axial location of the nozzle will be between the origin of the $35°$ half-angle flame contour and the position shown. The position of the central nozzle also can be used to regulate the small amount of air flow into the central tube. This air flow will be determined on the basis of smoke and coking considerations during tests mentioned above.

DILUTION AIR

Approximately 10 percent of the combustor air flow was admitted through radial holes downstream of the sections shown (Figure 8-1). Therefore, the gas temperature downstream of the last swirler was approximately $60°C$ higher than the T_{OB} of the burner. The absolute temperature level was below the temperature range where thermal NO_x reactions take place, but CO and smoke burnout might be influenced by the delayed introduction of the dilution air. Also, the profile factor can be influenced by adding 10 percent dilution air. Testing the full-size burner should supply the information needed to decide whether or not to use additional dilution air.

WALL STRUCTURE

The scaled version of the burner was built with the assumption that the high velocity, radially thick layers of 345°C combustion air will provide thermal protection for the MASB walls. The total testing time of about 35 hours is too short to predict the expected life of the combustor. During the full-scale tests, a decision can be made about the need for additional protection against overheating. Based on the experience so far, the design (Figure 8-2) does not show wall cooling devices. This arrangement must be confirmed by additional tests.

Section 9
CONCLUSIONS AND RECOMMENDATIONS

The program to develop fuel-flexible combustors with low emissions has
been successful in meeting its objectives. Four candidates for further
development have been tested and shown capable of meeting EPA require-
ments for ERBS fuel. Specific conclusions include:

- Rich/lean staged combustion has been shown to be successful in
  meeting EPA limits on ERBS fuel and also on fuels containing
  FBN.

- Catalytic combustion has been shown to be capable of meeting
  the program goal of ultra-low NOx emissions on ERBS fuel.

- Two combustors, the multiannular swirl burner and the Rolls
  Royce burner, are considered viable choices for near-term
  development due to their all-metal construction.

- Further development of the modular and catalytic burners will
  depend on accompanying ceramic materials development programs.

Based on these conclusions and the summary of test results, the following
recommendations are made for future development:

- Although the Rolls-Royce Combustor (Configuration 11) meets
  emissions goals on ERBS fuel, it is not recommended for further
  study because it is sufficiently defined for commercial develop-
  ment.

- The Lean Catalytic Burner will not be considered because other
  programs in progress are covering catalytic burners.

- Two configurations should be selected for full size development:
  the Rich/Lean Modular Combustor (Configuration 1/2) and the
  Multiannular Swirl Burner (Configuration 9).

- As the main objective of the first phase of the program was to
  screen various combustor concepts rather than to develop and

optimize them, some subscale work is recommended preceding and parallel to the design and testing of the full scale burner.

- The objectives of the subscale work on the Rich/Lean Modular Combustor will be:

  - To investigate whether the low NOx level of the ceramic components can be achieved with cooled metal structures

  - Establish the load ramping sequence for the metal version of the burner

  - Optimize the quench section and choose between Configurations 1 and 2

- The objectives of the subscale work on the Multiannular Swirl Burner will be

  - Optimize the swirl distribution to achieve minimum NOx smoke and pressure drop

  - Investigate the possible advantages of the radial dilution section over a dilution air supply integral with the rest of the burner

  - Minimize the secondary atomizing air flow and eliminate it if stability can be retained without it

  - Find optimal nozzle design (possibly eliminate primary atomizing air)

  - Find maximum throughput compatible with emissions performance to minimize the number of combustors

- Incorporate results of Phase IA (Low/Medium Btu Gas Tests) into Phase I (liquid fuel) designs, possibly develop multi-fuel capability for the same hardware

REPORT DOCUMENTATION PAGE

Public reporting burden for this collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection of information, including suggestions for reducing this burden, to Washington Headquarters Services, Directorate for Information Operations and Reports, 1215 Jefferson Davis Highway, Suite 1204, Arlington, VA 22202-4302, and to the Office of Management and Budget, Paperwork Reduction Project (0704-0188), Washington, DC 20503.

1. AGENCY USE ONLY (Leave blank)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	October 1981	Final Contractor Report

4. TITLE AND SUBTITLE

Low NO_x Heavy Fuel Combustor Concept Program
Phase I: Combustion Technology Generation

5. FUNDING NUMBERS

WU–None
DEN3–146

6. AUTHOR(S)

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7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

Westinghouse Electric Corporation
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P.O. Box 251
Concordville, Pennsylvania 19331

8. PERFORMING ORGANIZATION REPORT NUMBER

E–14186

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Washington, DC 20546–0001
and
U.S. Department of Energy, Office of Coal Utilization
Washington, DC 20545

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA CR–165482
DOE/NASA/0146–1

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified - Unlimited
Subject Category: 44

Available electronically at http://gltrs.grc.nasa.gov
This publication is available from the NASA Center for AeroSpace Information, 301–621–0390.

12b. DISTRIBUTION CODE

13. ABSTRACT (Maximum 200 words)

This report documents the results of a DOE/NASA sponsored project to conceptualize, design and demonstrate the viability of low emission (NO_x) gas turbine combustors for industrial and utility application. Thirteen different concepts were evolved and most were tested. Acceptable performance was demonstrated for four of the combustors using ERBS fuel and ultra-low NO_x emissions were obtained for lean catalytic combustion. Residual oil and coal derived liquids containing fuel bound nitrogen were also used at test fuels, and it was shown that staged rich/lean combustion was effective in minimizing the conversion of FBN to NO_x. The rich/lean concept was tested with both modular and integral combustors. While the ceramic lined modular configuration produced the best results, the advantages of the all-metal integral burners make them candidates for future development. An example of scaling the laboratory-sized combustor to a 100 MW size engine is included in the report as are recommendations for future work.

14. SUBJECT TERMS

Combustion; Emissions; Heavy fuel; Rich/lean combustion; Catalyst; Fuel bound nitrogen; Equivalence ratio

15. NUMBER OF PAGES

144

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102